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Lubricating Films and Machine Elements: Bearings, Slides, Ways, Gears, Couplings, Chains, Wire Rope

The elements of machines that require lubrication are bearings—plain, rolling element, slides, guides, and ways; gears; cylinders; flexible couplings, chains, cams, and cam followers; and wire ropes. These elements have fitted or formed surfaces that move with respect to each other by sliding, rolling, and advancing and receding, or by combinations of these motions. If actual contact between surfaces occurs, high frictional forces leading to high temperatures, and possibly wear or failure, will result. Therefore, the elements are lubricated to prevent or reduce the actual contact between surfaces.

Without lubrication, most machines would run for only a short time. With inadequate lubrication, excessive wear is usually the most serious consequence, since a point will be reached, usually after a short period of operation, when the machine elements cannot function and the machine must be taken out of service and repaired. Repair costs—material and labor—may be high, but lost production or lost availability of the machine may exact by far the greatest cost. With inadequate lubrication, even before failure of elements occurs, frictional forces between surfaces may be so great that drive motors will be overloaded or frictional power losses excessive. Finally, inadequately lubricated machines will not run efficiently, smoothly, or quietly.

Machine elements are lubricated by interposing and maintaining fluid films between moving surfaces. These films minimize actual contact between the surfaces, and because shearing occurs readily, the frictional force opposing motion of the surfaces is low.

I. TYPES OF LUBRICATING FILM

Lubrication films may be classified as follows:

1. *Fluid films* are thick enough that during normal operation they completely separate surfaces moving relative to each other. Fluid film lubrication includes hydrodynamic, elastohydrodynamic, hydrostatic, and squeeze types.

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2. *Thin films* are not thick enough to maintain complete separation of the surfaces all the time. Thin film lubrication includes mixed film and boundary lubrication.
3. *Solid films* are more or less permanently bonded onto the moving surfaces.

A. Fluid Films

Fluid film lubrication is the most desirable form of lubrication because during normal operation, the films are thick enough to completely separate the load-carrying surfaces. Thus friction is at a practical minimum, being due to shearing of the lubricant films alone, and wear does not occur because there is essentially no mechanical contact. Fluid films are formed in three ways:

1. Hydrodynamic and elastohydrodynamic films are formed by motion of lubricated surfaces through a convergent zone such that sufficient pressure is developed in the film to maintain separation of the surfaces.
2. Hydrostatic film is formed by pumping fluid under pressure between surfaces that may or may not be moving with respect to each other.
3. Squeeze films are formed by movement of lubricated surfaces toward each other.

Two types of hydrodynamic film lubrication are now recognized. In plain journal bearings and tilting pad or tapered land thrust bearings operating under correct lubrication conditions, thick hydrodynamic films are formed. These films are usually in excess of 0.001 in. (25 μm) thick. In hydrodynamic applications, the loads are low, and the areas over which the loads are distributed are relatively large; thus the amount of deformation of the load-carrying area is not great enough to significantly alter that area. Pressures in full hydrodynamic lubrication films range from a few psi to several thousand psi. Typical pressures found are in the range of 50–300 psi. Load-carrying surfaces of this type are often referred to as ''conforming,'' although it is obvious that in the case of tapered land thrust bearings, for example, the surfaces do not conform in the normal concept of the word. However, the term is a convenient opposite for the term ''nonconforming,'' which quite accurately describes surfaces of the types at which elastohydrodynamic films are formed.

The surfaces of the balls in a ball bearing theoretically make contact with the raceways at points; rollers of roller bearing make contact with the raceways along lines; and meshing gear teeth bearing also make contact along lines. These types of surface are nonconforming. Under the pressure (30,000–400,000 psi) applied to these elements by load conditions through the lubricating film, the metals deform elastically, expanding the theoretical points or lines of contact into discrete areas. The oil is able to generate films at these extremely high pressures because of the characteristic of viscosity doubling for each 5000 psi increase in applied pressure. This is discussed later in this chapter.

Since a convergent zone exists immediately before these areas of contact, a lubricant will be drawn into the contact area and can form a hydrodynamic film. This type of film is referred to as an elastohydrodynamic film, the ''elasto'' part of the term referring to the requirement that elastic deformation of the surfaces occur before the film can be formed. This type of lubrication is called elastohydrodynamic lubrication (EHL: the acronym EHD is also used). EHL films are very thin, in the order of 10–50 μin. (0.25–1.25 μm) thick. However, even with these thin films, complete separation of the contacting surfaces can be obtained.

Any material that will flow at the shear stresses available in the system may be used for fluid film lubrication. In most applications, petroleum-derived lubricating oils or

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synthetic fluids are used. There are some applications for greases. Some materials not usually considered to be lubricants, such as liquid metals, water, and gases, are also used in some applications. The following discussions are concerned mainly with fluid film lubrication with oils and greases, since some rather complex special considerations are involved when other materials are used.

1. Thick Hydrodynamic Films

The formation of a thick hydrodynamic fluid film that will separate two surfaces and support a load can be described with reference to Figure 8.1. The two surfaces are submerged in a lubricating fluid. As the upper surface moves, internal friction in the fluid causes it to be drawn into the space between the surfaces. The force drawing the fluid into space *A* is equal to the force tending to draw it out; but since the cross-sectional area at the outlet section is smaller than the inlet, the flow of fluid is restricted at the outlet. The moving surface tries to ''compress'' the fluid to force it through this restricted section, with the result that the pressure in the fluid rises. This pressure rise tends to do the following:

Retard the flow of fluid into space *A*
Increase the flow at the restricted outlet section
Cause side leakage in the direction normal to the direction of motion

The most important effect, however, is that the pressure in this wedge-shaped film enables it to support a load without contact occurring between the two surfaces.

For the elements of Figure 8.1 to work as a bearing, the moving surface would have to be of infinite length. A more practical approach is to distribute a series of wedge-shaped sections around the circumference of a disk as shown in the inset. The moving surface

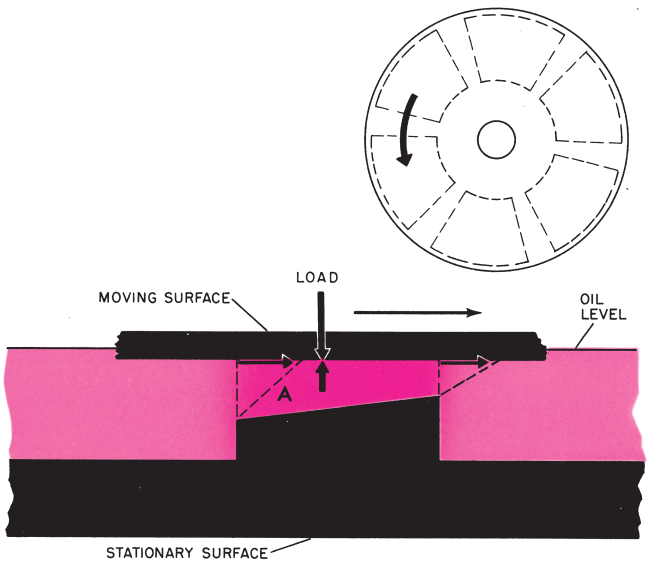

Figure 8.1 Converging wedge.

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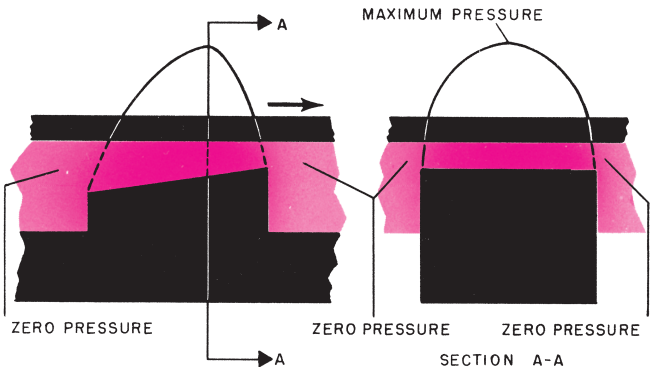

Figure 8.2 Pressure distribution in the wedge film: left, pressure distribution in the direction of motion; right, pressure distribution in the direction normal to motion.

then becomes a disk to which load can be applied, and the complete assembly can be used as a thrust bearing.

The pressure distribution in the wedge-shaped fluid film of Figure 8.1 is shown in Figure 8.2. In the direction of motion (left view), the pressure gradually rises more or less to a peak just ahead of the outlet section and then falls rapidly. In the direction normal to motion (right view), the pressure is at its maximum in the middle of the shoe, but because of side leakage, falls off rapidly toward each side.

In Figure 8.2, the taper of the shoe and the thickness of the fluid film are greatly exaggerated for illustrative purposes. In a practical bearing of this type, the amount of taper might be on the order of 0.002 in. (50 μm) in a length of 6 in. (150 mm), and the film thickness at the outlet would be on the order of 0.001 in. (25 μm). With proper design and fluid viscosity, there would be no contact between the disk and shoes when the speed is above a certain minimum during normal operation. The bearing would not be frictionless, however, since a force would have to be applied to the disk to overcome the resistance to shear of the fluid films.

The same principles outlined in Figure 8.2 can be used to generate a fluid film, which will lift and support a shaft or journal in a bearing. If the moving surface and tapered shoe of Figure 8.2 are ''rolled up,'' the journal and bearing of Figure 8.3 will result. A journal turning at a suitable speed will then draw fluid into the space between the journal and the bearing and develop a load-supporting film. This bearing is called a partial bearing and is suitable for some applications in which the load on the journal is always in one direction. However, bearings are more commonly of the full 360-degree type as shown in Figure 8.4 to provide restraint in the event of load variations and to permit easier enclosure and sealing.

The stages in the formation of a hydrodynamic film in a full journal bearing are illustrated in Figure 8.4. First the machine is at rest with the oil shut off, and the oil has leaked from the clearance space. Metal-to-metal contact exists between the journal and bearing. When the machine has been started (left) and the oil supply turned on, filling the clearance space, the shaft begins to rotate counterclockwise and friction is momentarily high, so the shaft tends to climb the left-hand side of the bearing. As it does this, it rolls onto a thicker oil film so that friction is reduced and the tendency to climb is balanced by the tendency to slip back. As the journal gains speed (center), drawing more oil through

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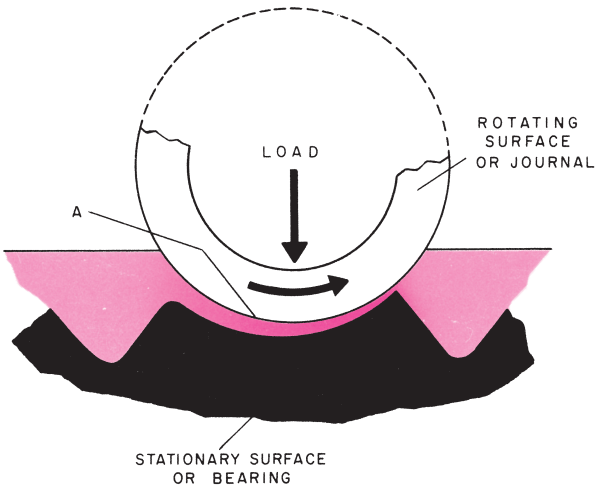

Figure 8.3 Partial journal bearing.

the wedge-shaped space between it and the bearing, pressure is developed in the fluid in the lower left portion of the bearing that lifts the journal and pushes it to the right. At full speed (right), the converging wedge has moved under the journal, which is supported on a relatively thick oil film with its minimum thickness at the lower right, on the opposite side to the starting position shown (left). Under steady conditions, the upward force developed in the oil film just equals the total downward load, and the journal is supported in the slightly eccentric position shown. Variations in speed, load, and oil viscosity will cause changes in this eccentricity, and also in the minimum film thickness. The pressures developed in the oil film are illustrated in Figure 8.5.

Thick hydrodynamic film lubrication is used in journal and thrust bearings for many applications. Characteristics and lubricant requirements of various applications are discussed more fully in later sections.

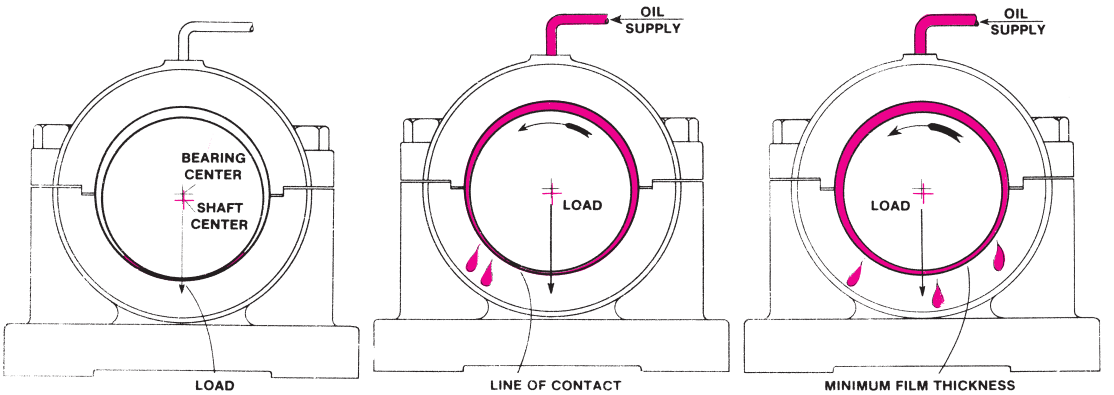

Figure 8.4 Development of hydrodynamic film in a full journal bearing with downward load.

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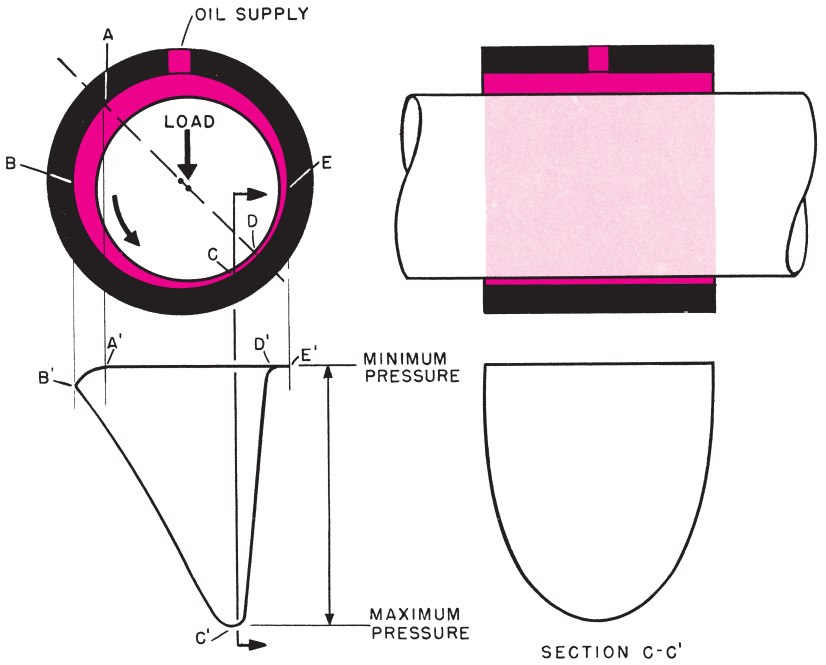

Figure 8.5 Pressure distribution in a full journal bearing. In the left view from *C,* the point of minimum film thickness, to *D,* the pressure drops rapidly to the minimum, which may be atmospheric, slightly above atmospheric, or slightly below atmospheric depending on such factors as bearing dimensions, speed, load, oil viscosity, and supply pressure. From *A* to *C* the pressure increases as the clearance decreases. Pressure distribution along the line of minimum film thickness is shown at the right. From a maximum value near the center, it drops rapidly to zero at the ends because of end leakage.

2. Elastohydrodynamic Lubricating (EHL) Films

In the lubrication of EHL contacting surfaces, such as antifriction bearings, the viscosity of the oil must increase as the pressure on the oil increases (Figure 8.6) to establish a supporting film at the very high pressures in the contact area. Thus, as the lubricant is carried into the convergent zone approaching the contact area, elastic deformation of two surfaces, such as a ball and its raceway or a roller and its raceway, results from the pressure of the lubricant. As the viscosity increases, the pressure is further increased. This hydrodynamic pressure developed in the lubricant is sufficient to separate the surfaces at the leading edge of the contact area. As the lubricant is drawn into the contact area, the pressure on it rises further, increasing its viscosity. This high viscosity and the short time required for the lubricant to be carried through the contact area prevent the lubricant from escaping, and thus separation of the surfaces can be achieved.

As noted, EHL films are very thin, on the order of about 10–50 μin. (0.25–1.25 μm). The thickness of the film increases as the speed is increased, the lubricant viscosity is increased, the load is decreased, or the geometric conformity of the mating surfaces is improved. Load has comparatively little effect on the film thickness because at the pres-

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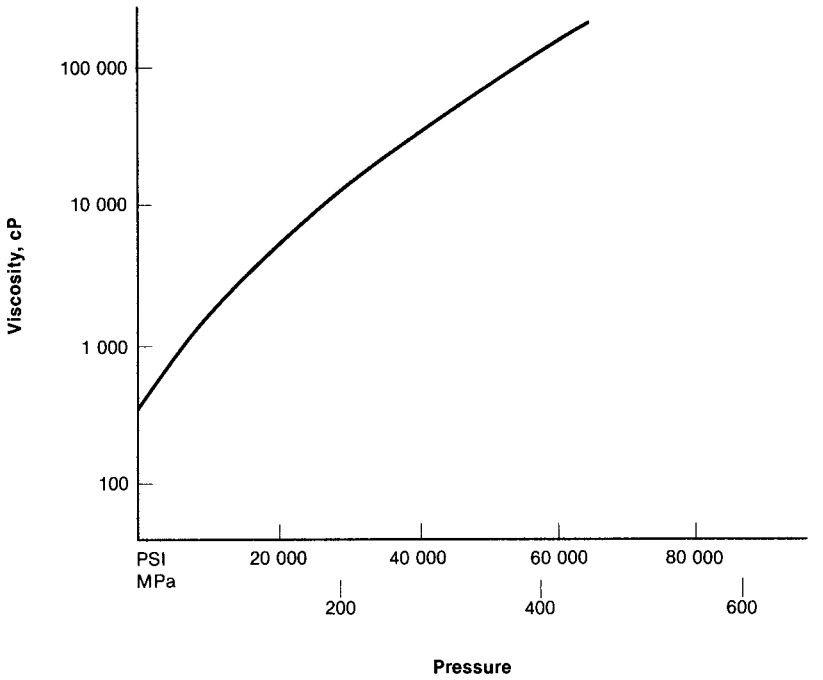

Figure 8.6 Effect of pressure on viscosity.

sures in the contact area, the oil film is actually more rigid than the metal surfaces. Thus an increase in load, for example, mainly has the effect of deforming the metal surfaces more and increasing the area of contact rather than decreasing the film thickness.

If the surfaces shown in Figure 8.7 were rougher, that is, if the asperities were higher, the opposing surface asperities would be more likely to make contact through the film. Thus the relationship between the roughness of the surfaces and the film thickness is important in the consideration of elastohydrodynamic lubrication. This has led to the introduction of the quantity specific film thickness λ in EHL considerations. The specific film thickness is the ratio of the film thickness h to the composite roughness σ of the two surfaces:

$$\lambda = h/\sigma$$

The composite roughness of the two surfaces can be calculated as the square root of the sum of the squares of the individual surface roughness:

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$$

where σ_1 and σ_2 are the root-mean-square (rms) roughness values of the two surfaces.*

* A number of calculations are given in the literature for the composite roughness of using centerline average (CLA) values. The relationship between rms and CLA values is $\sigma_{RMS} = 1.3\sigma_{CLA}$.

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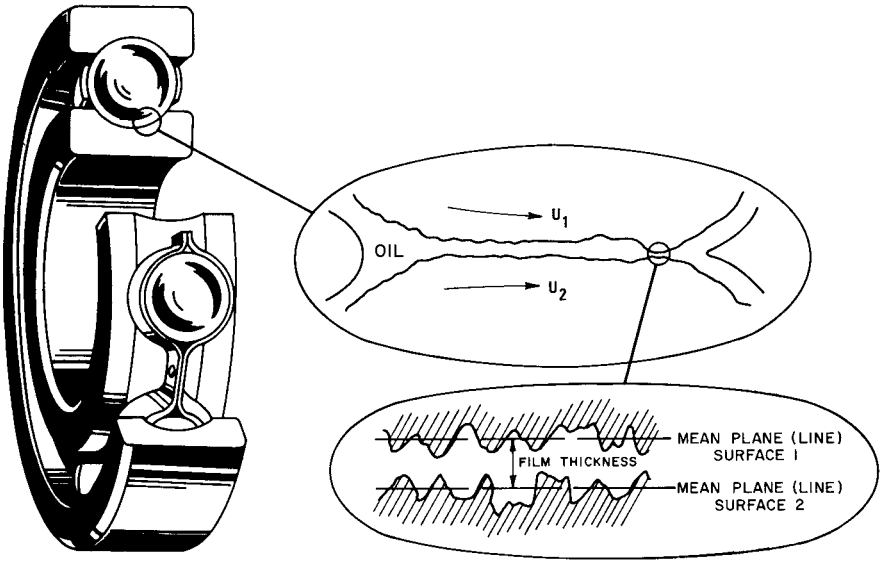

Figure 8.7 Schematic of contact area between ball and inner race.

The specific film thickness is related to the ability of the lubricating film to prevent or minimize wear or scuffing. Figure 8.8 shows the effect of λ on the L_{10} fatigue life† of a series of cylindrical roller bearings. For values of λ of approximately 3–4, the L_{10} fatigue life is relatively constant, so is essentially independent of the oil film thickness. This region is labeled ''full EHL'' and corresponds to full fluid film conditions that exist in thick hydrodynamic film lubrication. In the region labeled ''partial EHL,'' the film becomes progressively thinner, and more and more surface asperities penetrate the film.

Fatigue life is decreased accordingly. This region is one of mixed film lubrication. When λ is below about 1, boundary lubrication prevails and the surfaces are in contact nearly all the time. Fatigue failure is generally rapid unless lubricants with effective anti-wear additives are used. For λ values above 4, full film conditions exist; however, the thicker oil films can result in higher bearing temperatures and lower overall efficiency.

These ranges for λ are considered to be applicable to rolling element bearings. As will be discussed later, somewhat lower values of the calculated safe minimum film thickness may be applicable to gears.

Increased understanding of EHL film characteristics and the use of specific film thickness has led to improved understanding of the lubricant function in many applications where unit loads are high, as well as to improved methods of selecting lubricants for such applications.

3. Hydrostatic Films

In hydrostatic film lubrication, the pressure in the fluid film that lifts and supports the load is provided from an external source. Thus, relative motion between opposing surfaces

† The L_{10} fatigue life is the average time to failure of 10% of a group of identical bearings operating under the same conditions.

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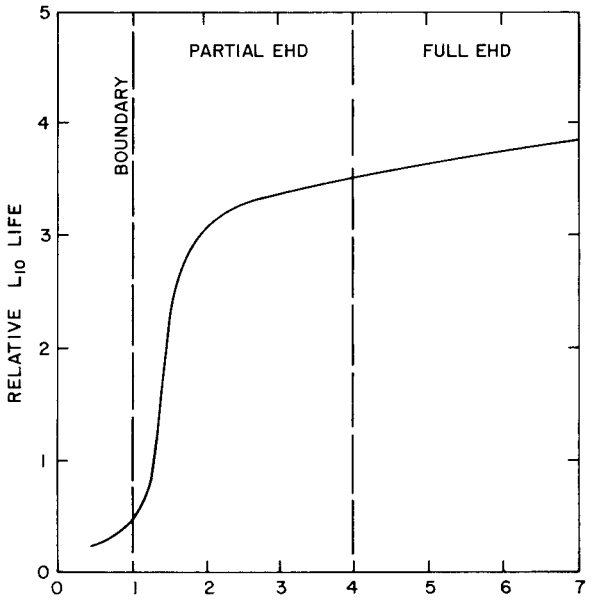

Figure 8.8 L_{10} fatigue life relative to book value as a function of specific film thickness; the data are for a series of cylindrical roller bearings.

is not required to create and maintain the fluid film. The principle is used in plain and flat bearings of various types, and it offers such advantages as low friction at very low speeds or, when there is no relative motion, more accurate centering of a journal in its bearing, and freedom from stick–slip effects.

The simplest type of hydrostatic bearing is illustrated in Figure 8.9. Oil under pressure is supplied to the recess or pocket. If the supply pressure is sufficient, the load will be lifted and floated on a fluid film. At equilibrium conditions, the pressure across the pad will vary approximately as shown. The total force developed by the pressure in the pocket and across the lands will be such that the total upward force is equal to the applied load.

The clearance space and the oil film thickness will be such that all the oil supplied to the bearing can flow through the clearance spaces under the pressure conditions prevailing.

4. Squeeze Films

As discussed in connection with EHL films (Section 1.A.2), as the applied pressure on an oil increases, the viscosity of the oil increases. This fact contributed to the formation of what are called squeeze films.

The principle of the squeeze film is shown in Figure 8.10, where application of a load causes plate *A* to move toward stationary plate *B*. As pressure develops in the oil layer, the oil starts to flow away from the area. However, the increase of pressure also causes an increase in the oil viscosity, so the oil cannot escape as rapidly and a heavy load can be supported for a short time. Sooner or later, if load continues to be applied, most of the oil will flow or be forced from between the surfaces, and metal-to-metal

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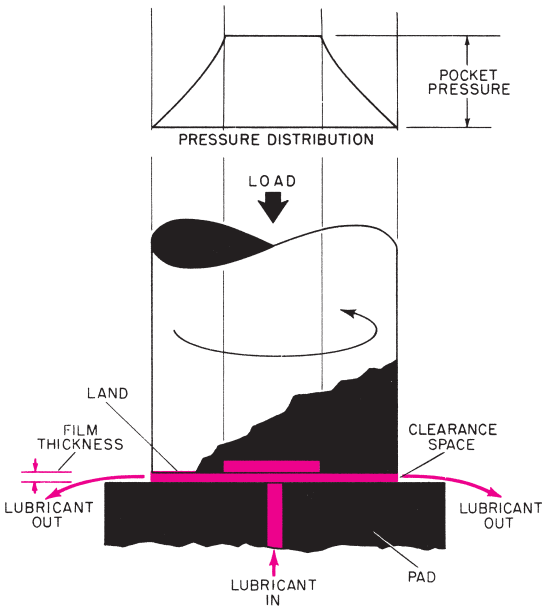

Figure 8.9 Simple hydrostatic thrust bearing.

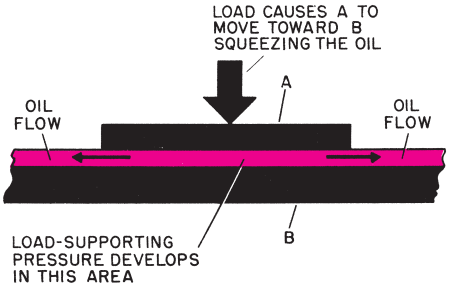

Figure 8.10 Squeeze film principle.

contact will occur. For short periods, however, such a lubricating film can support very heavy loads.

One application in which squeeze films are formed is found in piston pin bushings. At the left in Figure 8.11, the load is downward on the pin and the squeeze film develops at the bottom. Before the film is squeezed so thin that contact can occur, the load reverses (right view) and the squeeze film develops at the top. The bearing oscillates with respect to the pin, but this motion probably does not contribute much to film formation by hydrodynamic action. Nevertheless, bearings of this type have high load-carrying capacity.

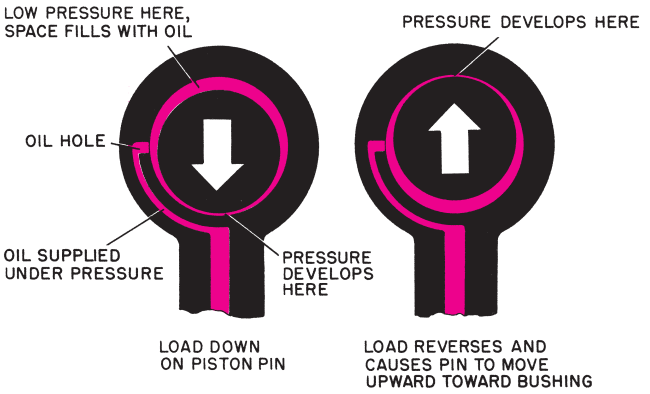

Figure 8.11 Squeeze film in piston pin bushing.

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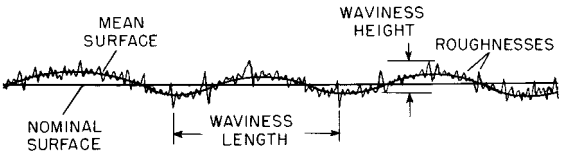

Figure 8.12 Magnified surface profile.

B. Thin Films

A copious, continuous supply of lubricant is necessary to maintain fluid films. In many cases, it is not practical or possible to provide such an amount of lubricant to machine elements. In other cases (e.g., during starting of a hydrodynamic film bearing), loads and speeds are such that fluid film cannot be maintained. Under these conditions, lubrication is by the so-called thin films.

When surfaces run together under thin film conditions, enough oil is often present to permit part of the load to be carried by fluid films and part by contact between surfaces. This condition is often called *mixed film lubrication.* With less oil present, or with higher loads, a point is reached at which fluid oil plays less of a role in separating the metal surfaces. This condition is often called *boundary lubrication.*

1. Nature of Surfaces

Machined surfaces are not smooth but are wavy because of inherent characteristics of the machine tools used. Minute roughnesses—''hills and valleys''—are superimposed on these waves, somewhat as shown in Figure 8.12. The surfaces are not clean even when freshly machined, but are coated with incidental films (moisture, oxides, cutting fluids, rust preventives) and, in service, with lubricating films. These films must be pierced or rubbed away before contact between clean surfaces can occur.

2. Surface Contact

When rubbing contact is made between the surface peaks, known as asperities, a number of actions take place, as shown in Figure 8.13, which represents a highly magnified contact area of a bearing and journal. These actions are as follows.

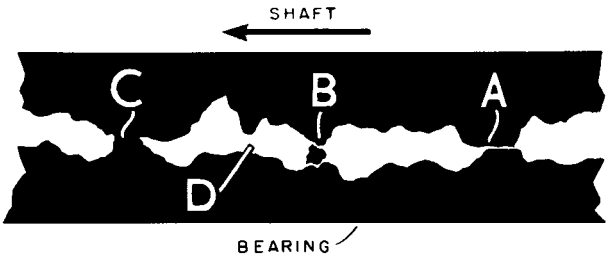

Figure 8.13 Actions involved in surface contacts.

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1. There is a heavy rubbing as at *A;* surface films are sheared and elastic or plastic deformation occurs. Real (vs. apparent) areas of contact are extremely small, and unit stresses will be very high.
2. The harder shaft material plows through the softer bearing material *B,* breaking off wear particles and creating new roughness.
3. Some areas *C* are rubbed or sheared clean, and the clean surfaces weld together. The minute welds break immediately as motion continues, but depending on the strength of the welds, the break may occur at another section, with the result that metal is transferred from one member to the other. New roughnesses are formed, some to be plowed off to form wear particles.

These actions account for friction and wear under boundary lubrication conditions. Under mixed film conditions, the force necessary to shear the oil films *D* also is part of the frictional force. However, the presence of partial oil films reduces friction and wear that result from the other actions.

The actions involved in rubbing contact result in the development of very high temperatures in the minute areas of contact. These local temperatures, which are of short duration, are known as flash temperatures. With light loads and low speeds, the small total amount of heat is conducted away through the surfaces without excessive rise in the general temperatures of the surfaces. However, with higher loads or speeds, or both, the number of contacts in a given area may increase to the point at which the heat cannot be dissipated fast enough, and the surfaces then may run hot.

Under *mixed film* conditions, the moving surfaces are kept apart by very thin films of lubricant. These thin films are subjected to high shear conditions, which can lead to increased temperatures, thereby reducing the effective viscosity of the oil. Quite often, owing to the increase in temperature, excessive loads, shock loads, slower speeds, stop and start operation or break-in of new components, the fluid films collapse and contact of the surfaces occurs. The collapse of the *mixed film* conditions result in *boundary lubrication.*

To help reduce the friction and wear under boundary conditions, it may be necessary to increase the viscosity or to use an oil with additives that provide special properties. These additives include film strength agents and extreme pressure (EP) additives. The film strength agents, sometimes referred to as oiliness or lubricity additives, attach themselves to the metal surfaces by polar attraction. This results in tenacious films that help separate the metal surfaces under moderate boundary conditions. Mild antiwear additives also help under moderate conditions by chemically attaching themselves to the metal surfaces.

As boundary lubrication conditions become more severe, increased additive activity may be needed to prevent accelerated wear or destruction of the surfaces. EP additives composed of chemically active ingredients such as phosphorus, sulfur, chlorine, or boron are needed for the more severe conditions. The EP additives work by chemically reacting with the rolling and sliding surfaces to form a strong surface film. The surfaces in motion then tend to wear away the surface film rather than the metal surfaces. Even under these conditions, shearing of the high spots (asperities) coming into contact may still occur. The EP additives also reduce the potential for welding or fusing of these sheared surfaces. It is important to recognize that EP additives need high temperatures for activation and therefore do not provide any advantages in applications characterized by low temperatures or surfaces separated by full fluid films.

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C. Solid or Dry Films

In many applications, oils or greases cannot be used because of difficulties in applying them, sealing problems, or other factors such as environmental conditions. A number of more or less permanently bonded lubricating films have been developed to reduce friction and wear in applications of this type. The solid or dry film lubricants reduce the effective surface roughness. They attach themselves to the metal surfaces through rubbing action or by chemical reaction. This effect is dependent on load, temperature, and types of material.

The simplest type of solid lubricating film is formed when a low friction, solid lubricant such as molybdenum disulfide is suspended in a carrier and applied more or less in the manner of a normal lubricant. The carrier may be a volatile solvent, grease, or any of several other types of material. When the carrier is squeezed or evaporates from the surfaces, a layer of molybdenum disulfide remains to provide lubrication.

Solid lubricants are also bonded to rubbing surfaces with resins of various types, which cure to form tightly adhering coatings with good frictional properties. In the case of some plastic bearings, the solid lubricant is sometimes incorporated in the plastic. This also occurs with some sintered metal bearings. During operation, some of the solid lubricant may then be transferred to form a lubricating coating on the mating surface.

In addition to molybdenum disulfide, polytetrafluoroethylene (PTFE), graphite, polyethylene, and a number of other materials are used to form solid films. In some cases, combinations of several materials, each contributing some characteristics to the film, are used.

II. PLAIN BEARINGS

The simplest types of plain bearing are shown in Figure 8.14. The bearing may be only a hole in a block (left); it may be split to facilitate assembly (center); or in some cases, where the load to be carried is always in one direction, the bearing may consist of only a segment of a block (right). The part of the shaft within a bearing is called the journal, and plain bearings are often called journal bearings.

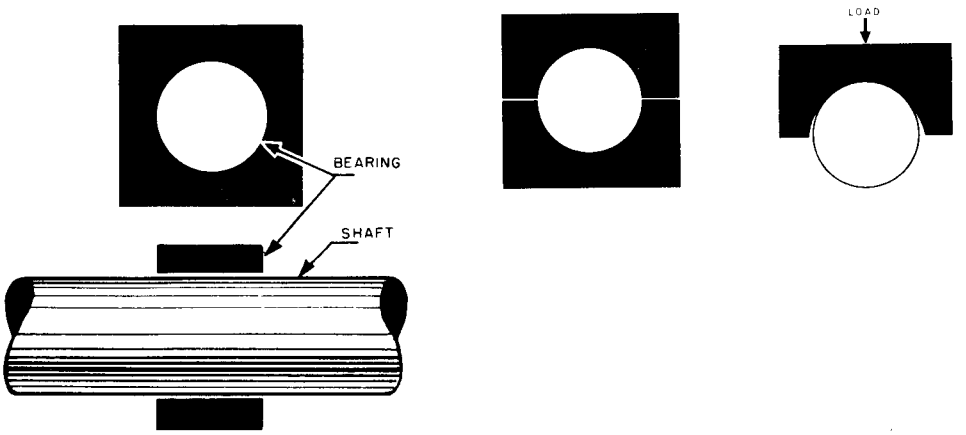

Figure 8.14 Elementary plain bearings.

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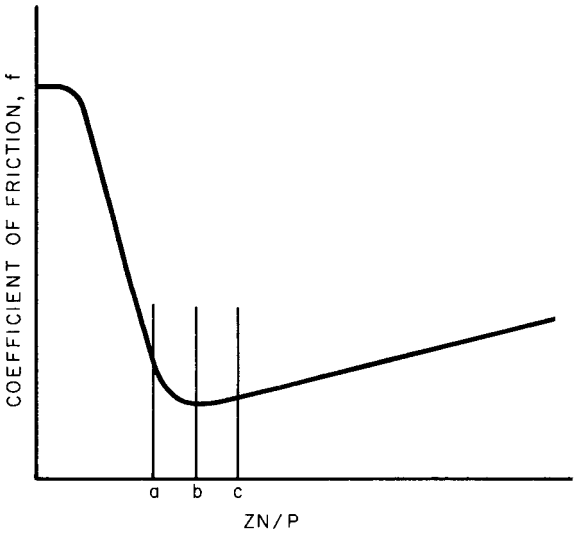

Figure 8.15 Effect of viscosity, speed, and load on bearing friction. For each hydrodynamic film bearing, there is a characteristic relation, such as that shown, between the coefficient of friction and the factor combining viscosity (Z), speed (N), and load (P).

Plain bearings are designed for either fluid film lubrication or thin film lubrication. Most fluid film bearings are designed for hydrodynamic lubrication, but increasing numbers of bearings for special applications are being designed for hydrostatic lubrication.

A. Hydrodynamic Lubrication

The primary requirement for hydrodynamic lubrication is that oil of correct viscosity and sufficient quantity be present at all times to flood the clearance spaces.

The oil wedge formed in a hydrodynamic bearing is a function of speed, load, and oil viscosity. Under fluid film conditions, an increase in viscosity or speed increases the oil film thickness and the coefficient of friction, while an increase in load decreases them. The separate consideration of these effects presents a complex picture that is simplified by combining viscosity *Z,* speed *N,* and unit load *P,* into a single dimensionless factor called the *ZN*/*P* factor.* Although no simple equation can be offered that expresses the coefficient of friction in terms of *ZN*/*P,* the relationship can be shown by a curve such as that in Fig 8-15. A similar type curve could be developed experimentally for any fluid film bearing.

In Figure 8.15, in the zone to the right of *c,* fluid film lubrication exists. To the left of *a,* boundary lubrication exists. In this latter zone, conditions are such that a full fluid

* The expression *ZN*/*P* is dimensionless when all quantities are in consistent units: for example, *Z* in poises, *N* in revolutions per second, and *P* in dynes per square centimeter; or *Z* in pascal-seconds, *N* in revolutions per second (reciprocal seconds), and *P* in pascals; or *Z* in reyns, *N* in revolutions per second, and *P* in pounds-force per square inch.

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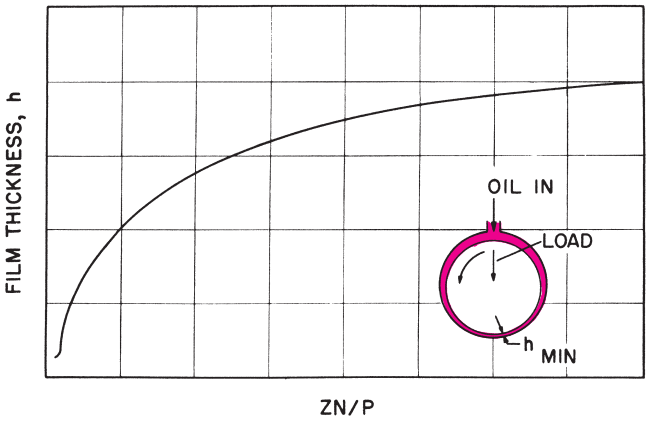

Figure 8.16 Effect of viscosity, speed, and load on film thickness.

film cannot be formed, some metallic friction and wear commonly occur, and very high coefficients of friction may be reached.

The portion of the curve between points *a* and *c* is a mixed film zone including the minimum value of *f* corresponding to the *ZN*/*P* value indicated by *b.* From the point of view of low friction, it would be desirable to operate with *ZN*/*P* between *b* and *c,* but in this zone any slight disturbance such as a momentary shock load or reduction in speed might result in film rupture. Consequently, good practice is to design with a reasonable factor of safety so that the operating value of *ZN*/*P* is in the zone to the right of *c.** The ratio of the operating *ZN*/*P* to the value of *ZN*/*P* for the minimum coefficient of friction (point *b*) is called the bearing safety factor. Common practice is to use a bearing safety factor on the order of 5.

In an operating bearing, if it becomes necessary to increase the speed, *ZN*/*P* will increase and it may be necessary to decrease the oil viscosity to keep *ZN*/*P* and the coefficient of friction in the design range. An increase in load will result in a decrease in *ZN*/*P,* and it may be necessary to increase the oil viscosity to keep *ZN*/*P* and the coefficient of friction in the design range.

Film thickness can be related to *ZN*/*P* in the manner shown in Figure 8.16. The curve is typical of large, uniformly loaded, medium speed bearings such as are used in steam turbines. In general, film thickness increases if *ZN*/*P* is increased—for example, if the load is reduced while the oil viscosity and journal speed remain constant. With a proper bearing safety factor, the film thickness will be such that normal variation in speed, load, and oil viscosity will not result in the reduction of film thickness to the point at which metal-to-metal contact will occur.

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* Equations, procedures, and data for plain bearing design and performance calculations are available in many technical papers and books. Among the latter are the following: *Bearing Design and Application,* Wilcock and Booser, McGraw-Hill, *Theory and Practice of Lubrication for Engineers,* Fuller, John Wiley & Sons; *Analysis and Lubrication of Bearings,* Shaw and Mack, McGraw-Hill.

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The work done against fluid friction results in power loss, and the energy involved is converted to heat. Most of the heat is usually carried away by the lubricating oil, but some of it is dissipated by radiation or conduction from the bearing or journal. The normalized operating temperature is the result of a balance between the heat generated, overcoming fluid friction, and the total heat removal. Certain oils, such as some synthetics, have naturally lower frictional characteristics, which can reduce power requirements.

The effect of increasing temperature is to decrease oil viscosity. The reduction in viscosity results in a lower ZN/P and coefficient of friction (provided boundary or mixed film lubrication conditions do not exist). Also, less work is required to overcome fluid friction, less heat is developed, and the temperature tends to decrease. This has a stabilizing influence on bearing temperatures.

In general, if excessive temperatures develop even though load, speed, and oil viscosity are within the correct range, it may be that there is insufficient oil flow for proper cooling. It may then be necessary to provide extra grooving or increase the clearance in order to increase the flow of oil through the bearing.

1. Grease Lubrication

While the grease in a rolling element bearing acts as a two-component system in which the soap serves as a sponge reservoir for the fluid lubricant, greases in plain bearings behave like homogeneous mixtures with unique flow properties. These flow properties are described by the apparent viscosity (see Chapter 4), that is, the observed viscosity under each particular set of shear conditions. As the rate of shear is increased, the apparent viscosity decreases and, at high shear rates, it approaches the viscosity of the fluid lubricant used in the formulation. In many plain bearings, the shear rate in the direction of rotation is high enough to cause the apparent viscosity of a grease to be in the same general range as the viscosities of lubricating oils normally used for hydrodynamic lubrication. As a result, fluid film formation can occur with grease, and it is now believed that some grease-lubricated plain bearings operate on fluid films, at least part of the time. In addition, hydrodynamic film bearings designed for grease lubrication are used in some applications.

The pressure distribution in a grease-lubricated hydrodynamic film bearing is similar to that in an oil-lubricated bearing (Figure 8.5). However, toward the ends of the bearing, because of reduced pressure in the film, the shear stress is lower, the apparent viscosity of the grease remains high, and end leakage is lower. As a result, high pressures are maintained farther out toward the ends of the bearing; moreover, the average pressure in the film is higher, and the maximum pressure is correspondingly lower. The minimum film thickness for the same bearing load and speed will be greater. The coefficient of friction may be equal or less than that with an equivalent oil-lubricated bearing, depending on such factors as the type of grease used and the viscosity of the oil component in the grease.

Fluid film bearings lubricated with grease have some advantages compared to those lubricated with oil. As a result of the lower end leakage, the amount of lubricant required to be fed to the bearing is less, so grease-lubricated bearings can be supplied by an all-loss system with either a slow, continuous feed, or a timed, intermittent feed in conjunction with adequate reservoir capacity in the grooves of the bearing.

When a grease-lubricated bearing is shut down for a period of time with the flow of lubricant shut off, the grease usually does not drain or squeeze out completely. Some grease remains on the bearing surfaces, and thus a fluid film can be established almost immediately when the bearing is restarted. Starting torque and wear during starting may

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be greatly reduced. During shutdown periods, retained grease also acts as a seal to exclude dirt, dust, water, and other environmental contaminants, and to protect bearing surfaces from rust and corrosion. If the grease provides a lower coefficient of friction, power consumption during operation will also be lower.

When grease lubrication is used for fluid film bearings, the cooling is not as efficient as the cooling obtained from oils. This disadvantage may be partially offset if the coefficient of friction is lower with a grease; if speeds or loads are high, however, it may be a limitation.

B. Hydrostatic Lubrication

In a hydrostatic bearing, the oil feed system used must be such that the pressure available, when distributed across the pocket and land surfaces, is sufficient to support the maximum bearing load that may be applied. The system must also be designed to provide an equilibrium condition for loads below the maximum. Three types of lubricant supply are used to accomplish this-constant volume system, constant pressure system with flow restrictor, and constant pressure system with flow control valve.

1. Constant Volume System

In the first type of system, the pump delivers a constant volume of oil at whatever pressure is necessary to force that volume through the system. That is, if the backpressure increases, the pump pressure automatically increases sufficiently to maintain the flow rate. In most cases, the volume delivered by the pump actually decreases somewhat as the pressure increases, but this has relatively little effect on the way the system operates.

A constant volume system must have adequate pressure capability to support any applied load. Referring to Figure 8.9, when the pump is turned on, oil will flow into the pocket and the pressure will increase until the load is lifted sufficiently to establish a clearance space through which the volume of oil flowing in the system will be discharged. The clearance space and oil film thickness will be functions of the volume of flow in the system, the viscosity of the lubricant, and the applied load.

If the load is then increased, the clearance space and film thickness will decrease, and the pump pressure will have to increase to permit the discharge of the same volume of oil through the reduced clearance space. Only small changes in clearance space and film thickness accompany fairly large variations in load, so the bearing is said to be very ''stiff.''

The disadvantage of the constant volume system is that it does not compensate for variations in the point of application of the load in multiple pocket bearings. In the two-pocket bearings of Figure 8.17, using a constant volume system, if the load is shifted to the right, the runner will tend to tilt. This will decrease the clearance at the right-hand land and increase the clearance at the left-hand land. Oil can then flow more freely out of the left pocket, the pressure in the system will decrease, and the load will sink until metallic contact might occur at the right side. This problem can be compensated for with either of the following systems.

2. Constant Pressure System with Flow Restrictor

A constant pressure system requires an accumulator or manifold to maintain the pressure at a relatively constant value. If this constant pressure is applied to the pockets of the bearing (Figure 8.17) through flow restrictors, such as capillaries or orifices, a compensat-

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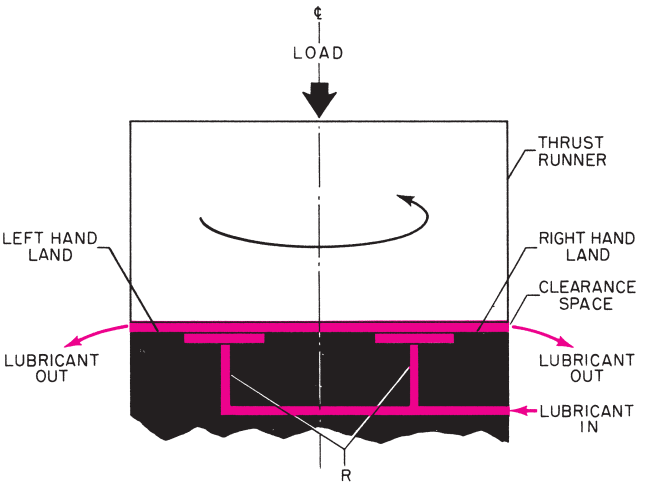

Figure 8.17 Two-pocket hydrostatic thrust bearing. When flow restrictors are used, they are installed at the locations marked *R*.

ing force will be developed when the runner tends to tilt. The pressure drop across a flow restrictor increases as the flow through it increases, and decreases as the flow decreases. Thus, the pressure in the pocket, being the difference between the system pressure and the pressure drop across the restrictor, decreases as flow increases and increases as flow decreases. With this system, if an off-center load causes the runner to tilt toward the right, the clearance at the right-hand land will decrease, the flow to that pocket will decrease, the pressure drop across the restrictor will decrease, and the pocket pressure will increase. The opposite effect will occur in the other pocket. This will tend to lift the load over the right-hand pocket and let it sink over the left-hand pocket, restoring the runner to a more or less coaxial position.

In single-pocket bearings, a constant pressure system with a flow restrictor is less stiff than in a constant volume system. As the load is increased with the constant pressure system, the pocket pressure must increase to support it. This requires that the oil flow decrease to reduce the pressure drop across the flow restrictor. With less flow and higher pressure, a considerably smaller clearance space is required to discharge the oil. However, the compensating feature results in wide use of constant pressure systems in both thrust and journal bearings with multiple pockets.

3. Constant Pressure with Flow Control Valve

If a flow control valve is used as the restrictor in a constant pressure system, the system becomes essentially constant pressure and constant volume. The bearing will have the stiffness of a constant volume system as well as the compensating features of the constant pressure system.

4. Hydrostatic Bearing Applications

One of the more common applications of the hydrostatic principle is in ‘‘oil lifts’’ for starting heavy rotating machines, such as steam turbines, large motors in steel mills, and

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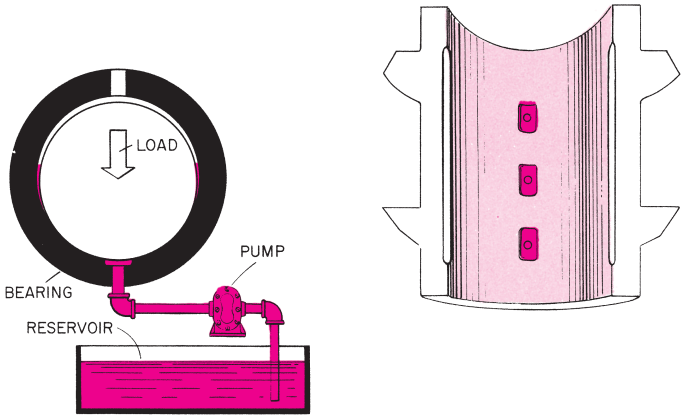

Figure 8.18 Hydrostatic lift; the view at the right shows one type of shallow pocket through which the oil pressure can be applied.

rotary ball and rod mills. Hydrostatic lifts for plain bearings are also used for turning gear operation during start-up and cooldown periods of large steam and gas turbines, where the turbine rotors are rotated at speeds too slow to establish hydrodynamic films. Because metal-to-metal contact exists between the journal and the bearings when the journal is at rest, extremely high torque may be required to start rotation, and damage to the bearings may occur. By feeding oil under pressure into pockets machined in the bottoms of the bearings, the journal can be lifted and floated on fluid films (Figure 8.18). The pockets are generally kept small to prevent serious interference with the hydrodynamic film capacity of the bearings. When the journal reaches a speed sufficient to create hydrodynamic films, the external pressure can be turned off and the bearings will continue to operate in a hydrodynamic manner. The reverse procedure may be used during shutdown.

The low friction characteristics of hydrostatic film bearings at low speeds are being used in a variety of ways. One application is in ''frictionless'' mounts or pivots for dynamometers. Another is in the bearings for tracking telescopes where the relative motion is extremely slow but must be completely free of stick–slip effects. Increasingly, the hydrostatic principle is being applied to the guides and ways of large machine tools, particularly when extremely precise movement and location of the ways is required.

The characteristic of controlled film thickness of hydrostatic film bearings is being used in high speed applications such as machine tool spindles for high precision work. Spindles of this type are equipped with multiple pocket bearings with a constant pressure system and a flow restrictor for each pocket. With this arrangement, any change in the lateral loading on the spindle as a result of a change in the cutting operation is automatically compensated for by changes in the pressures in the individual pockets. Lateral movement of the spindle is thus minimized, and very accurate control of the centering of the spindle in the bearings can be achieved.

C. Thin Film Lubrication

Many bearings are designed to operate on restricted lubricant feeds as the most practical and economic approach. The lubricant supplied to the bearings gradually leaks away and

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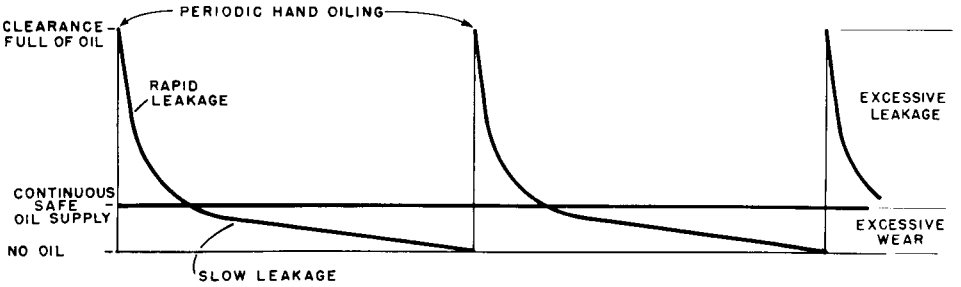

Figure 8.19 Hand oiling: the condition of ''feast or famine'' that is always present with periodic hand oiling is compared with the safe continuous supply of oil that is closely approximated by devices that feed oil frequently in small quantities.

is not reused; thus this type of lubrication is generally referred to as ''all-loss'' lubrication. Because of the restricted supply of lubricant, these bearings operate on thin lubricating films, either of the mixed film or boundary type. The simplest type of all-loss lubrication is hand oiling (see Figure 8.19). Hand oiling results in flooded clearances immediately after lubrication. This condition may permit formation of fluid films for a brief period of time; however, the oil quickly leaks away to an amount less than that considered to be acceptable for safe operation. In short, the bearing passes through the regime of mixed film lubrication and operates much of the time under boundary conditions.

A closer approach to maintaining a safe oil supply may be accomplished with application devices such as wick feed oilers, drop feed cups, waste-packed cups, bottle oilers, and central dispensing systems such as force feed lubricators or oil mist systems. These devices supply oil on either a slow, constant basis or at regular, short intervals. With greases, leakage is not as serious a problem, but the use of centralized lubrication systems will provide a more uniform lubricant supply than grease gun application (see Chapter 8).

Even with regular application of small amounts of lubricant, thin film bearings require proper design and installation, as well as proper lubricant selection to control wear and provide satisfactory service life.

1. Wearing In of Thin Film Bearings

In a new bearing, the journal normally will make contact with the bearing over a fairly narrow area (Figure 8.20, left). Generally lighter loads should be carried by such a bearing under thin film conditions, since unit loads beyond the ability of the oil film to prevent metallic contacts would probably exist. Under favorable conditions, wear will occur, but it will have the effect of widening the contact area (Figure 8.20, right) until the load is distributed over a region so large that wear becomes practically negligible. New plain bearings generally are supplied with a thin ''flashing'' (approximately 0.0005 in.) of a softer material to help facilitate break-in. Under unfavorable conditions, this initial wear may be so rapid that bearing failure occurs.

Large bearings are often fitted prior to operation by hand scraping, or by counterboring the loaded area to the radius of the journal. Fitting of this type can be done only when

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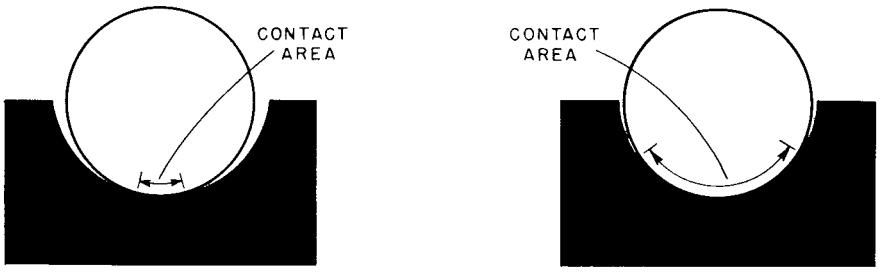

Figure 8.20 Contact area before and after wearing-in of plain thin film bearing.

the direction of loading is known and is constant. After fitting, the area of contact should be in the range of 90–120 degrees.

D. Mechanical Factors

In plain bearings, there are several mechanical factors that affect lubrication. These include the following:

1. Ratio of the length of the bearing to its diameter
2. Projected area of the bearing
3. Clearance between journal and bearing
4. Bearing materials
5. Surface finish
6. Grooving (where used)

1. Length-to-Diameter Ratio

The diameter of a journal is determined by mechanical requirements, which involve the torque transmitted, shaft strength, and shaft rigidity. The bearing length is governed by the load to be carried, the space available, and the operating characteristics of the particular type of machine, especially with regard to retention of bearing alignment and shaft flexing. More and more, space limitations are becoming extremely important in the determination of bearing length.

As a general rule, the shorter the bearing, the higher end leakage will be, and the more difficult it will be to develop load-supporting lubricant films. On the other hand, with too long a bearing, metal-to-metal contact, high friction, high temperatures, and wear may occur at the bearing ends if there is even moderate misalignment or shaft deflection under load. Length-to-diameter (L/D) ratios of as much as 4 have been used in the past, but the trend for many years has been to lower ratios. Most fluid film bearings now have L/D values of 1.5 or less; connecting rod bearings for internal combustion engines may have L/D of 0.75 or less. Thin film bearings have L/D ratios in about the same range.

2. Projected Area

The axial length of a bearing multiplied by its diameter is called the projected area, and it is common to express unit loads in force per unit of projected area. Established practices pertaining to unit loads on this basis vary considerably for different classes of machinery

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and different bearing materials, ranging from as low as 15 psi (103 kPa) for lightly loaded line shafting to as high as 5000 psi (24.5 MPa) or more for internal combustion engine crankpins and wristpins. Most industrial bearings carrying constant loads—as in turbines, centrifugal pumps, and electrical machinery—fall in the range of 50–300 psi (345–2700 kPa), with most under 200 psi (1380 kPa). Heavier loads are encountered in bearings of reciprocating machinery and in other bearings subject to varying or shock loads. Peak hydraulic pressures within the oil films (Figure 8.5) are usually three to four times these unit loads based on projected area.

To achieve optimum life in plain bearings, full film (hydrodynamic) lubrication is necessary. Other contributing factors to bearing life are speeds, loads, temperatures, and the compressive strength of the bearing materials. If the compressive strength of the materials used for metallic plain bearings is known, a good rule of thumb to achieve good life is that bearing loads not exceed 33% of the compressive strength of the materials. The limiting load and speed conditions can be expressed as a factor *PV,* with *P* being the pressure on the bearing (psi) multiplied by the surface speed *V* of the shaft (ft/min). The *PV* factor varies by bearing design and materials used. Data on *PV* factors and compressive strengths of materials can be obtained from the bearing manufacturers or, if the materials used in the bearing are known, is readily available technical manuals.

3. Clearance

A full bearing must be slightly larger than its journal to permit assembly, to provide space for a lubricant film, and to accommodate thermal expansion and some degree of misalignment and shaft deflection. This clearance between journal and shaft is specified at room temperature.

One of the principal factors controlling the amount of clearance that must be allowed is the coefficient of thermal expansion of the bearing material. The higher the coefficient of thermal expansion, the more clearance must be allowed to prevent binding as the bearing warms up to operating temperature. Babbitt metals and bearing bronzes have the lowest coefficients of thermal expansion of common bearing materials. Clearances for these materials in general machine practice range from 0.1 to 0.2% of the shaft diameter (0.001–0.002 in. per inch of shaft diameter). Many precision bearings have less clearance than this, while a rough machine bearing may have more. Because of their higher coefficients of thermal expansion, aluminum bearings require somewhat more clearance than babbitt metals or bronzes, and some of the plastic bearing materials require considerably more, in some cases as much as 0.8% of the shaft diameter.

4. Bearing Materials

During normal operation of a fluid film lubricated bearing under constant load, the most important property required in the bearing material is adequate compressive strength for the hydraulic pressures developed in the fluid film. When cyclic loading is involved, as in reciprocating machines, the material should have adequate fatigue strength to operate without developing cracks or surface pits. With shock loading, the material should be of such ductility that neither extrusion nor crumbling occurs. Under boundary lubrication conditions, the material also requires the following:

1. Scoring resistance, requiring appreciable hardness and low shear strength
2. The ability to conform to shaft irregularities and misalignment
3. The ability to embed abrasive particles

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If operating temperatures are high, resistance to corrosion and softening may be important.

Although these properties are somewhat conflicting, numerous materials have been developed to obtain satisfactory bearings for the wide range of conditions encountered.

Plain bearing materials most often encountered in industrial machines are bronzes and babbitt metals. Suitable bronzes and babbitt metals are available for practically all conditions of speed, load, and operating temperature encountered in general practice. Steel and cast iron are used for a limited number of purposes, usually involving low speeds or shock loads. There has been considerable growth in the use of plastic and elastomeric materials such as nylon, thermoplastic polyesters, laminated phenolics, polytetrafluoroethylene, and rubber for bearings, particularly in applications where contamination of, or leakage from, oil-lubricated bearings might result in high maintenance costs or short bearing life. Some of these materials can be lubricated with water or water soluble oil emulsions in certain applications. Allowable unit loads for these bearing materials usually are lower, although in a number of cases, filled nylon bearings have been used as direct replacements for bronze bearings.

For internal combustion engines, babbitt metal bearings are made with a very thin layer of babbitt over a backing of copper and/or steel to increase the load carrying capacity. Even then, the loads may be greater than babbitts can handle, so a number of stronger bearing materials have been developed. Aluminum bearings are being used in some diesel and gas engine applications because of their longer potential life and greater resistance to acid attack. Because the aluminum is harder, it will not embed particles as well as the softer bearing materials, and therefore contamination is more critical. Engine bearings are usually fabricated in the form of precision inserts (Figure 8.21), which are interchangeable and require no hand fitting machining at installation.

Precision insert bearings, which are usually constructed of layers of different materials, provide the following:

- A thin surface layer (sometimes as little as 0.0003 in., 0.0075 mm) having good surface characteristics—such as low friction, scoring resistance, conformability, and resistance to corrosion
- A thicker layer (0.008–0.025 in., 0.2–0.6 mm) of bearing material having adequate compressive strength and hardness, suitable ductility, and good resistance to fatigue
- A still thicker (usually 0.05–0.125 in., 1.25–3.2 mm) back or shell of bronze or steel

Some of the more common combinations used with this type of construction are babbitt metal over leaded bronze over steel, lead alloy over copper-lead over steel, silver alloy over lead over steel, and tin over aluminum alloy over steel. These bearings all require smooth hardened journals, rigid shafts and minimum misalignment.

5. Surface Finish

Machined surfaces are never perfectly smooth. The peak-to-valley depth of roughness in machined surfaces ranges from about 160 μin. (4 μm) for carefully turned surfaces to about 60 μin. (1.5 μm) precision-ground surfaces. Finer finishes, approximately 10 μin. (0.25 μm), can be obtained by other commercial methods.

Finely finished surfaces would, in general, be damaged less than rough surfaces by the metal-to-metal contact that occurs under boundary lubrication conditions. However,

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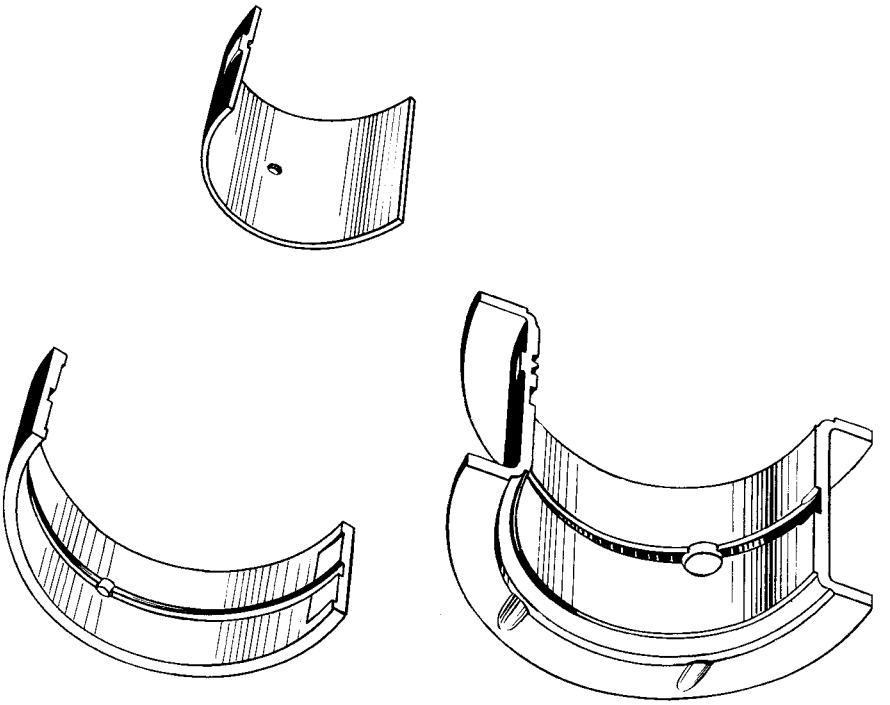

Figure 8.21 Precision insert bearings: typical main and connecting rod insert bearings used in internal combustion engines. Bearing at lower right is designed to carry thrust as well as radial loads.

some degree of ''wearing in'' of new bearing surfaces always occurs. New surfaces, which may be relatively rough, tend under favorable conditions and careful wearing in to become smoother. In some cases, certain degrees of initial roughness aid this wearing-in process (by holding lubricant in place), as long as loads are minimized for break-in.

Under fluid film conditions, the minimum safe film thickness is a function of the roughness of the surfaces; rougher surfaces requiring thicker films to prevent contact of surface asperities through the film. On the other hand, the finer the surface finish, the lower the minimum safe film thickness, and the less clearance is necessary. Since the film thickness decreases with increases in unit loading, if the minimum safe film thickness is lower as a result of finer surface finishes, the allowable unit loading is higher, all other factors being equal. Conversely, it can be said that bearings designed for high unit loads and small clearances must have finely finished surfaces. Tests show that fluid films may also be formed at a lower speed when starting up a bearing with a smooth finish than when starting one with a rough finish.

6. Grooving

In all plain bearings, some provision must be made to admit the lubricant to the bearing and distribute it over the load-carrying surfaces. Lubricant is generally admitted through an oil port or ports and then distributed by means of grooves cut in the bearing surface. The location of the supply port and the type of grooving used depend on several factors,

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including the type of supply system, the direction and type of load, and the requirements of the bearings. Certain basic principles apply to all cases.

(a) Grooving for Oil. The distribution of oil pressure in a typical fluid film bearing with steady load is shown in Figure 8.5. Usually, oil should be fed to a bearing of this type at a point in the no-load area where the oil pressure is low. When the shaft is horizontal and the steady load is downward, it is usually convenient to place the supply port at the top of the bearing, as shown.

Generally, grooves should not be extended into the load-carrying area of a fluid film bearing. Grooves in the load-carrying area provide an easy path for oil to flow away from the area. Oil pressure will be relieved and load-carrying capacity will be reduced. This effect for an axial and a circumferential groove is shown in Figures 8.22 and 8.23. However, to provide increased oil flow for better cooling in certain force-feed-lubricated bearings, it is sometimes necessary to extend the grooves through the load-carrying area. With variable load direction, it may also be necessary to extend the grooves through the load-carrying area. This is done in some precision insert bearings for internal combustion engines, mainly to increase cooling and oil distribution.

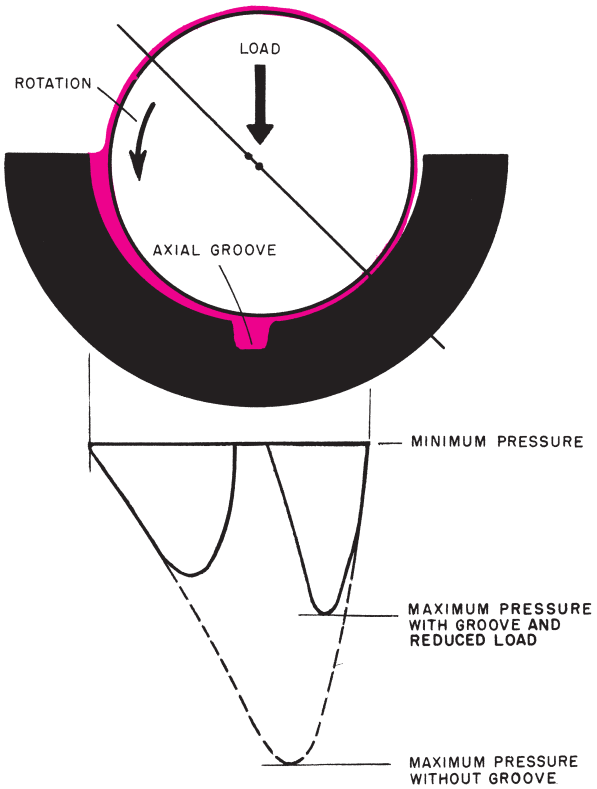

Figure 8.22 Axial groove reduces load-carrying capacity. An axial groove through the pressure area of a fluid film bearing provides an easy path for leakage and relief of oil pressure. Solid lines in the lower sketch represent the approximate pressure distribution when the groove is present; dashed line represents approximately what it would be without the groove.

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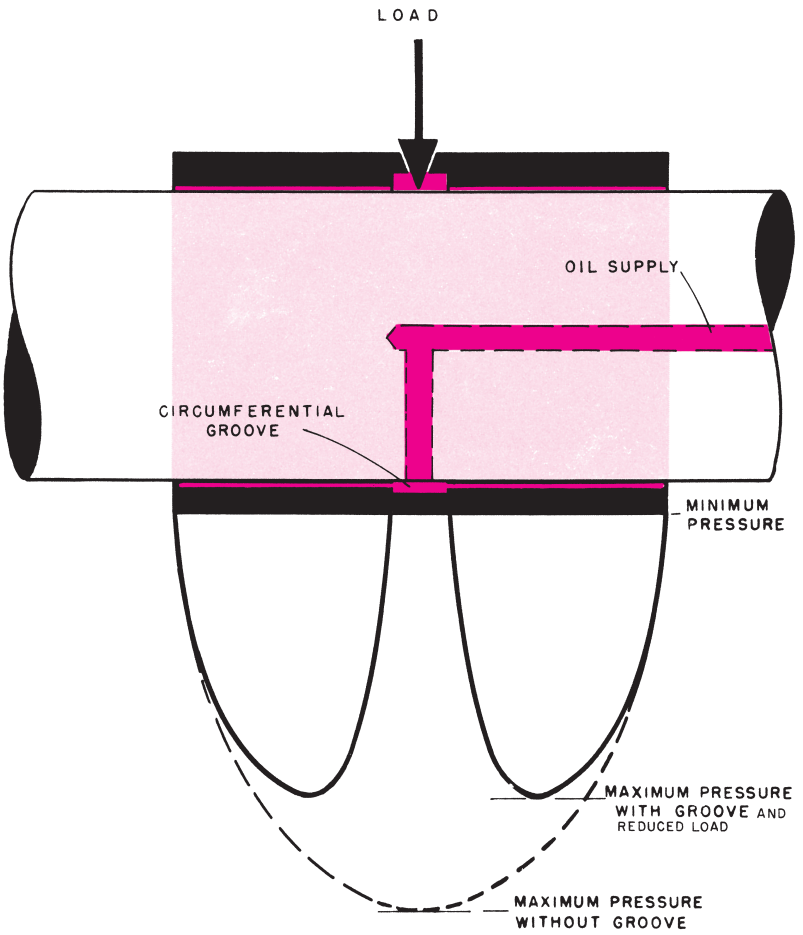

Figure 8.23 Circumferential groove reduces load-carrying capacity. As in Figure 8.22, solid lines show the pressure distribution with the groove present, dashed line shows what it would be without the groove.

With constant load direction, a single oil hole may be sufficient. To increase oil flow or improve distribution, an axial groove cut through the oil supply port (Figure 8.24) often is all that is required. Normally the groove should not extend to the ends of the bearing, since that would allow the oil to flow out the ends rather than being carried into the oil wedge. An exception to this is found when carefully sized ports or orifices are provided at the ends of the groove to permit increased flow for cooling purposes in certain high speed bearings. Also, in medium speed equipment, end bleeder ports frequently are provided to give a continuous flushing of dirt particles.

The grooving needed for distribution in a two-part bearing usually is formed by chamfering both halves at the parting line (Figure 8.25). Both sides are chamfered if it is necessary to provide for rotation in both directions. These chamfers should also stop short of the ends of the bearings. Frequently, oil inlet port and distribution grooves are combined with the chamfer at the split. If a top inlet port is used, an overshot feed groove may be machined in the upper half as shown in Figure 8.26.

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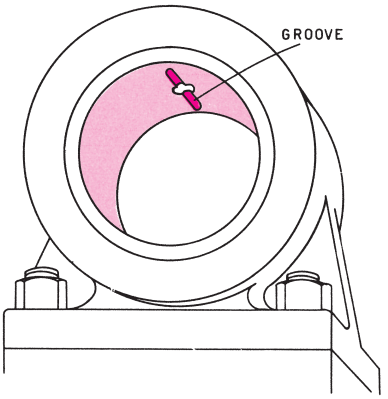

Figure 8.24 Axial distribution groove in one-part bearing.

If a stationary journal and a rotating bearing are used, oil may be fed through a port and axial groove in the journal. Again, the groove should be placed on the no-load side.

Where heavy thrust loads are to be carried, fluid film bearings of the tilting pad or tapered land type are often used. Tilting pad bearings require no grooving, since the oil can readily flow out around the pad mountings. Tapered land bearings require radial grooves located just ahead of the point where the oil wedge is formed. If thrust load is carried by one end face of a journal bearing, the axial groove or chamfers may be extended to the thrust end so that oil will flow directly to the thrust surfaces. The end of the bearing should be rounded or beveled to aid in the flow of oil between the end face and thrust collar or shoulder.

Circumferential grooves are sometimes cut near one or both ends of a bearing to collect end leakage and drain it to the sump or reservoir. This oil might otherwise flow along the shaft and leak through the shaft seals. When collection grooves are used, they mark the effective ends of the bearing.

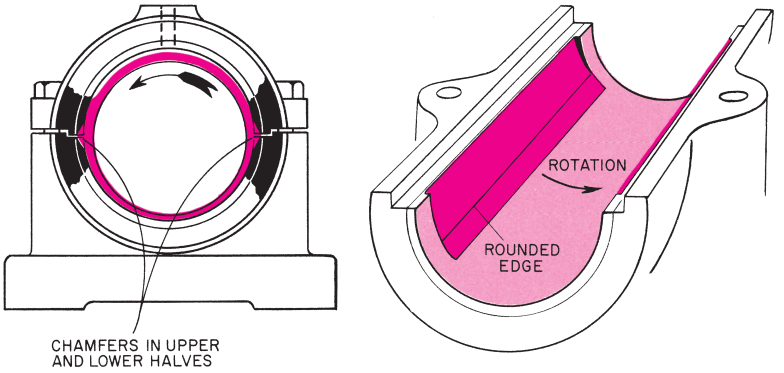

Figure 8.25 Distribution grooves in two-part bearing.

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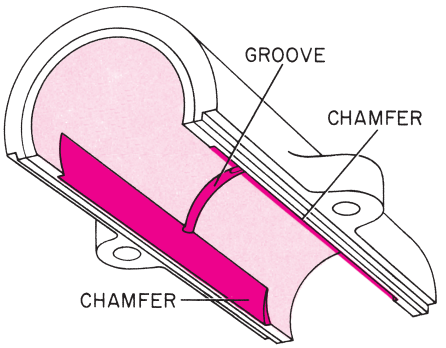

Figure 8.26 Overshot feed groove and chamfers.

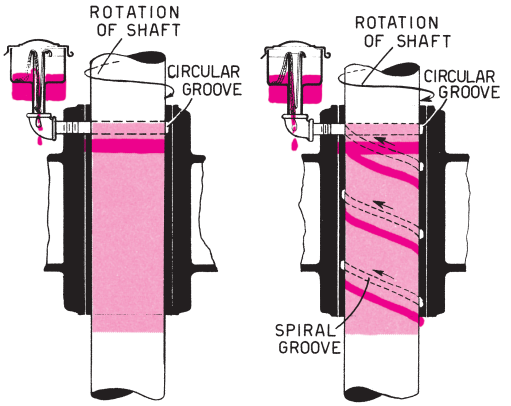

Figure 8.27 Grooving for vertical bearing.

Vertical shaft bearings often require only a single oil port in the upper half of the bearing in the no-load area. In general, the lower the supply pressure, the higher the port should be. Sometimes a circumferential groove may be added near the top of the bearing to improve distribution (Figure 8.27, left). If leakage from the bottom of the bearing is excessive, a spiral groove is sometimes cut in the bearing in the proper direction relative to shaft rotation so that oil will be pumped upward (Figure 8.27, right).

Increased oil flow to cool a hot running bearing can be obtained by simple forms of grooving. An axial groove on the no-load side, for example, will increase oil flow by three to four times compared to a single port alone. Circumferential grooves also increase oil flow, but not as much as an axial groove. They also have the disadvantage of reducing the load-carrying capacity of the bearing. Increased clearance often can be used in lightly loaded, high speed bearings to increase oil flow. When increased clearance might reduce

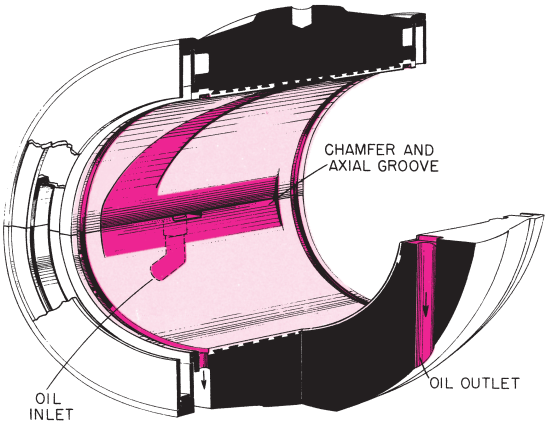

Figure 8.28 Grooving to increase oil flow for cooling: cutaway of a large turbine bearing shows a wide groove cut diagonally in the top (unloaded) half to permit a large flow of oil for cooling purposes. A relatively small part of the oil passing through this bearing would be needed for the fluid film.

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load-carrying capacity too much, extra grooving or a clearance relief in the unloaded portion can be used to increase oil flow for cooling (see Figure 8.28).

Where the direction of bearing load changes as in reciprocating machines, it is still essential that oil be fed into an unloaded or lightly loaded area. One way of doing this is with a circumferential groove. While this, in effect, divides the bearing into two shorter bearings of reduced total load-carrying capacity, it may be the most effective alternative. Also, it may be desirable to provide a path for oil flow to other bearings—for example, as in many internal combustion engines (Figure 8.29). An axial groove or chamfer may be used with a circumferential groove to improve oil distribution or to increase oil flow (Figure 8.30).

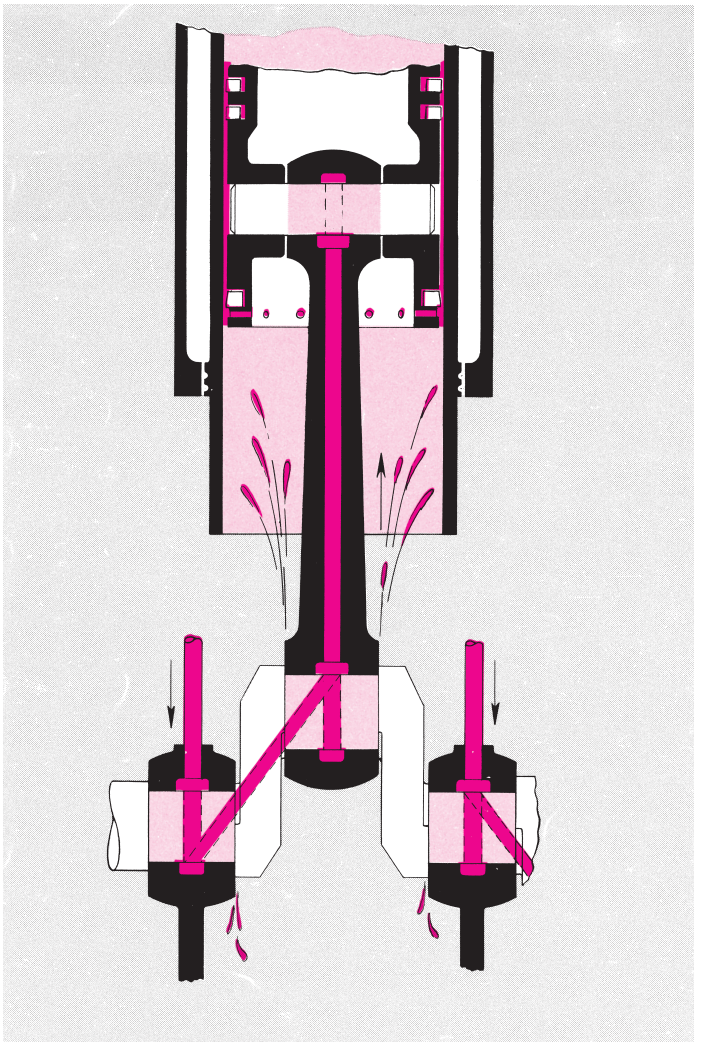

Figure 8.29 Circumferential grooves. In this circulation system an oil pump, driven from the crankshaft, takes oil from the crankcase sump and delivers it under pressure to the crankpin and wrist pin bearings through passages in the crankshaft and connecting rods.

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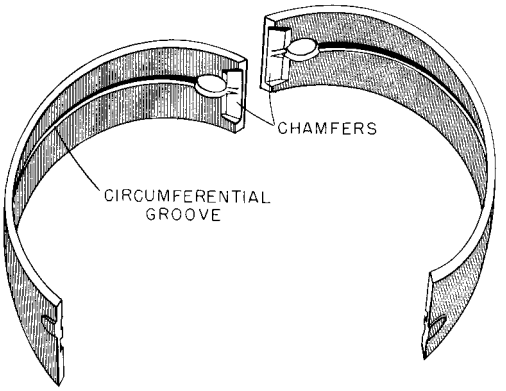

Figure 8.30 Circumferential groove and axial chamfer. An axial groove (also called a spreader groove) is often used with a circumferential groove in precision bearings.

Ring-oiled bearings are a somewhat special case with regard to grooving. When the direction of load is fixed and is toward the bottom of the bearing, a simple axial groove connecting with the ring slot (Figure 8.31) is adequate. In a two-part bearing, the axial groove may be formed by chamfering the bearing halves at the parting line. However, if loads are sideways because of belt pull or gear reaction, this type of groove can be blocked,

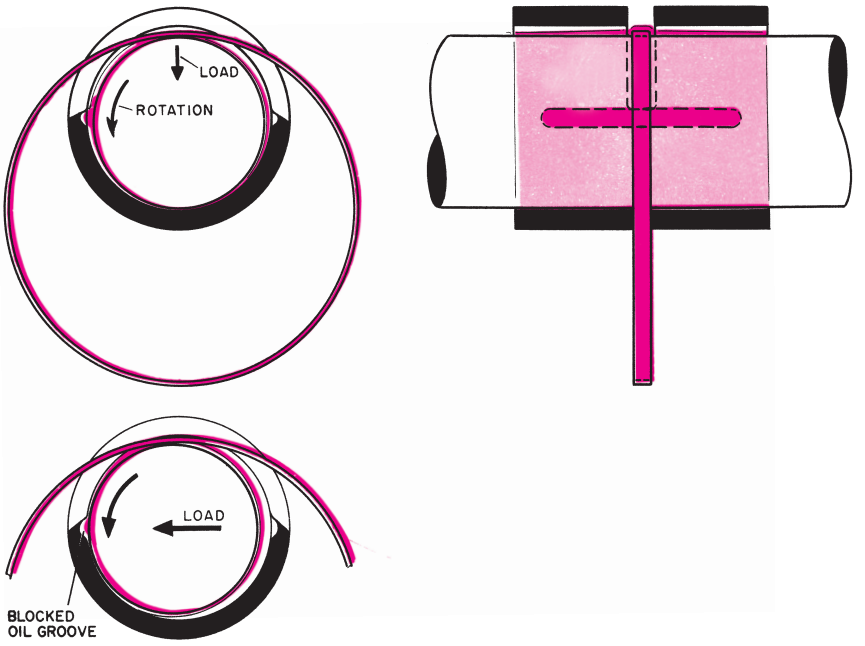

Figure 8.31 Ring-oiled bearing, downward load. A side thrust on the shaft can cause blockage of this type of grooving (lower left).

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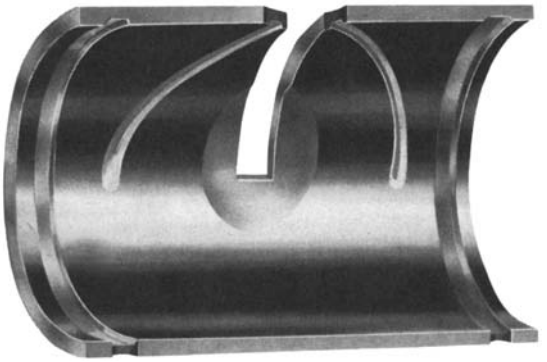

Figure 8.32 Section of ring-oiled bearing with X grooves, which cross in the top of the bearing. The area around the end of the ring slot is relieved to aid in the distribution of oil.

as shown in Figure 8.31. To overcome this problem, grooving such as shown in Figures 8.32 and 8.33 is often used. Grooving of this type is used for electric motors, which may be belted or geared to the load; thus the motor manufacturer does not need to know the contemplated direction of loading and shaft rotation.

Grooving for thin film lubricated bearings generally follows the same pattern seen with for fluid film lubricated bearings. The restricted supply of oil to this type of bearing, however, may necessitate an additional reservoir capacity in the grooves, which in turn would have to be designed to retain and distribute the oil (Figures 8.34–8.36).

(b) Grooving to Grease. Grooving principles for grease are practically the same as for oil. Because grease has high resistance to flow at the low shear rates in the supply system, grooves need to be made wider and deeper than oil grooves. Annular grooves cut near

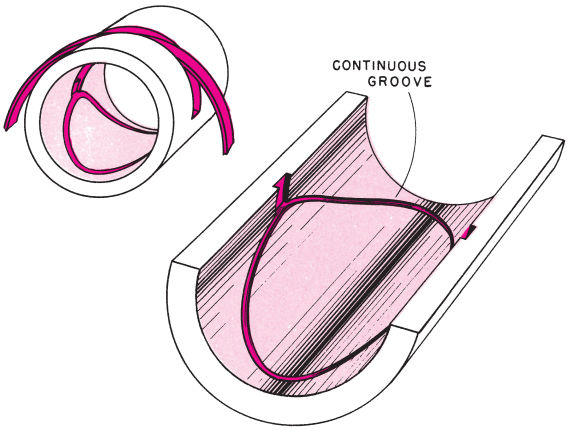

Figure 8.33 Ring-oiled bearing, continuous groove. With this pattern of grooving, oil feed cannot be blocked, and there is little interruption of load-carrying surface along any axial line.

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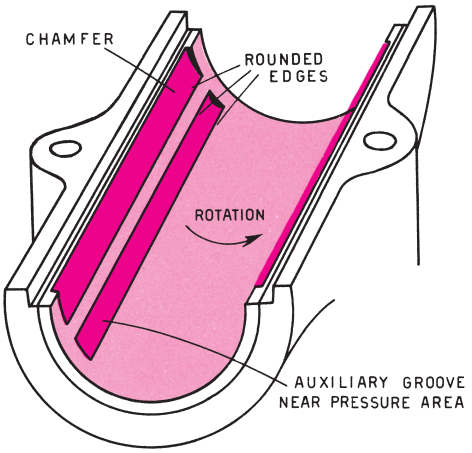

Figure 8.34 Auxiliary groove. When oil application is infrequent, extra storage capacity is sometimes needed in a bearing. This can be provided by means of an auxiliary groove cut just ahead of the load-carrying area.

the ends of the bearing, similar to oil collection grooves but without the drain hole, are often used to reduce grease leakage. Some of the grease forced into these grooves in the loaded zone flows back out in an unloaded area and enters the bearing for reuse.

E. Lubricant Selection

Lubricants for plain bearings must be carefully selected if the bearings are to give long service life with low friction and minimum power losses.

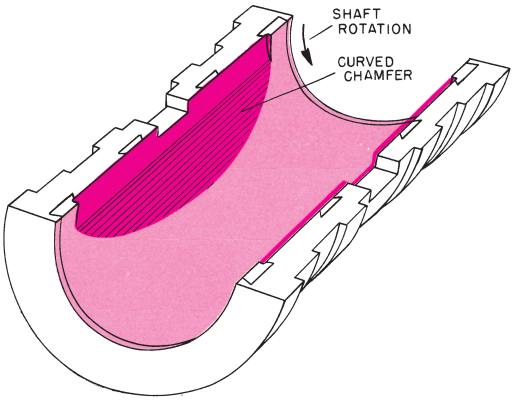

Figure 8.35 Curved chamfer. If clearance at the horizontal centerline is large, end leakage of oil can be reduced by using a curved chamfer, which tends to guide a restricted supply of oil toward the center of the bearing so that it will be drawn into the load-carrying area.

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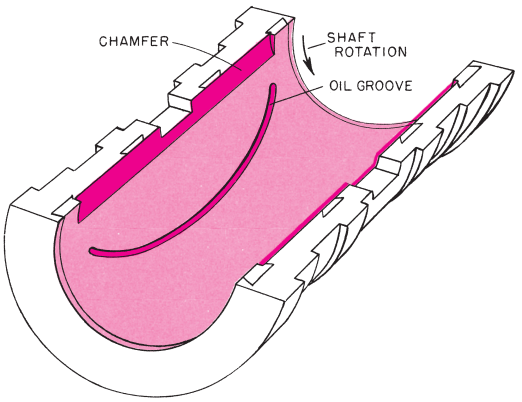

Figure 8.36 Straight chamfer and curved groove. The chamfer provides oil storage capacity and aids distribution. The curved groove catches oil flowing toward the bearing ends and guides it back toward the center of the bearing.

1. Oil Selection

Most of the lubrication systems for fluid film bearings are essentially circulation systems; thus, the oil is reused many times over long periods.

Moisture and other contaminants may enter the system. These conditions require that the oil have the following characteristics:

Good chemical stability to resist oxidation and deposit formation in long-term service
Ability to provide protection against rust and corrosion
Ability to separate readily from water to resist the formation of troublesome emulsions
Ability to resist foaming
Viscosity control in applications where dilution could occur

These properties are generally found in high quality circulation oils.

To select the proper viscosity of oil, three operating factors must be determined: speed, load, and operating temperature (including ambient temperatures at start-up). Speed (rpm) is usually readily determined. Bearing loads may be calculated from the total weight supported by the bearings divided by the total projected area of the bearings. For simplicity, in industrial-type applications, bearing loads of 200 psi (1380 kPa) can often be assumed for bearings operating under light to moderate loads with no shock loads, and 500 psi (3450 kPa) for heavily loaded bearings or where shock loads are present. Operating temperatures can be approximated by using the oil exit temperature of the hottest bearing.

When these three factors are known, the optimum viscosity of oil can be determined from a chart such as that shown in Figure 8.37. Since this chart gives the viscosity at the operating temperature, ASTM viscosity–temperature charts must then be used to determine the desired viscosity at one of the standard measuring temperatures (see discussion of viscosity in Chapter 3).

If bearings are to be started or operated at low temperatures, the pour point and low temperature fluidity of the oil selected must be considered. Synthesized hydrocarbon fluids (SHFs) are often used for low temperature bearing lubrication.

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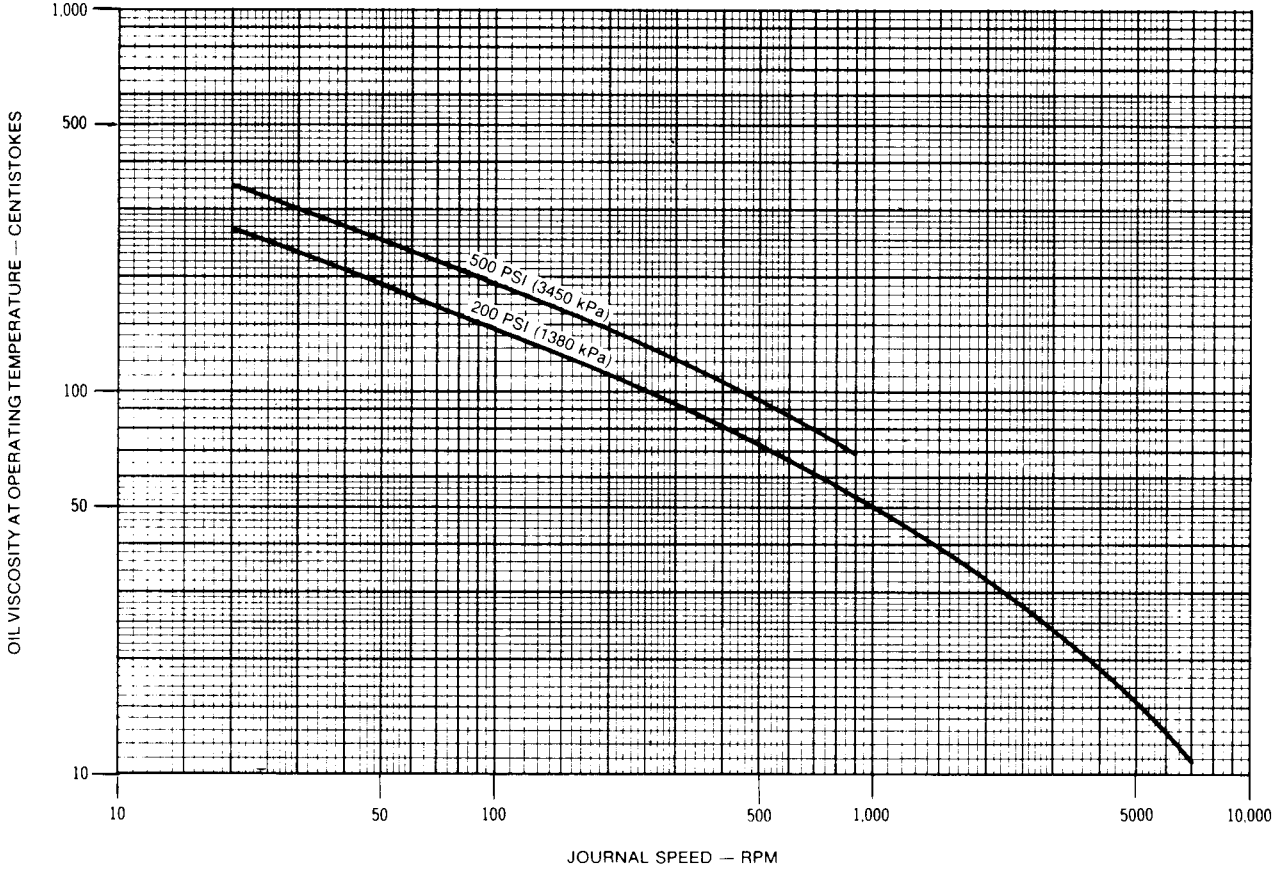

Figure 8.37 Oil viscosity chart for fluid film bearings. To determine the viscosity required, follow up from the appropriate journal speed to either of the lines, then to the left to read the oil viscosity at the operating temperature. The 200 psi line should be used for bearings operating under normal loads without shock loads, and the 500 psi line where loads are heavy or shock loads are present.

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In thin film lubricated bearings, the oil is normally not reused, so it is important that the oil selected be readily distributed and form tough, tenacious films on the bearing surfaces. Generally, oils with good film strength and friction reducing properties are preferred.

With thin film lubricated bearings, it may be assumed that loads are high enough for the bearings to operate under boundary conditions at least part of the time. Therefore, only speed and operating temperature must be determined to select the correct viscosity of oil.

Once speed and operating temperature are known, the oil viscosity may be obtained from a chart such as Figure 8.38. Again, the viscosity must be converted to a value at one of the standard reporting temperatures by means of viscosity–temperature charts. Ambient temperatures must also be considered, and so must the type of application device, since some of these devices are limited with respect to the maximum viscosity of oil they will dispense.

2. Grease Selection

The greases used for fluid film and thin film bearings are essentially similar. Enhanced film strength is probably required more frequently in thin film bearings, but where heavy, shock, or vibratory loads are encountered in fluid film bearings, this property is usually required.

Since the methods used to grease lubricate plain bearings are all-loss systems, the grease used is not subjected to long-term service that might cause oxidative breakdown. On the other hand, operating temperatures may be higher in grease-lubricated bearings than in comparable oil-lubricated bearings because of the poorer cooling ability of grease. Thus, there may be exposure to high temperatures while the grease is in the bearings. There is also severe mechanical shearing of the grease, particularly as it passes through the load-carrying zone, which may cause softening and lead to increased end leakage.

The method of application has considerable influence on both the type of grease and the consistency selected for plain bearings. With centralized lubrication systems, the grease must be a type suitable for dispensing through such systems. Where bearings must be lubricated at low ambient temperatures, softer greases or greases with good low temperature properties are usually needed, since they pump more easily at low temperatures.

These requirements generally dictate the use of greases with good mechanical stability, adequate dispensing and pumpability characteristics, corrosion protection properties, adequate film strength, and adequate high temperature characteristics for the operating temperatures. In addition, for fluid film bearings, the apparent viscosity of the grease must be sufficient to permit the formation of fluid films at the shear rates prevailing, but not so high that friction losses will be excessive.

III. ROLLING ELEMENT BEARINGS

The term ‘‘rolling element bearings’’ is used to describe that class of bearings in which the moving surface is separated from the stationary surface by elements such as balls, rollers, or needles that can roll in a controlled manner. These bearings are often referred to as ‘‘antifriction’’ bearings.

The essential parts of a rolling element bearing (Figure 8.39) are a stationary ring (cup or raceway), a rotating ring (cup or raceway), and a number of rolling elements. The inner ring fits the shaft or spindle, and the outer ring fits in a suitable housing. Shaped

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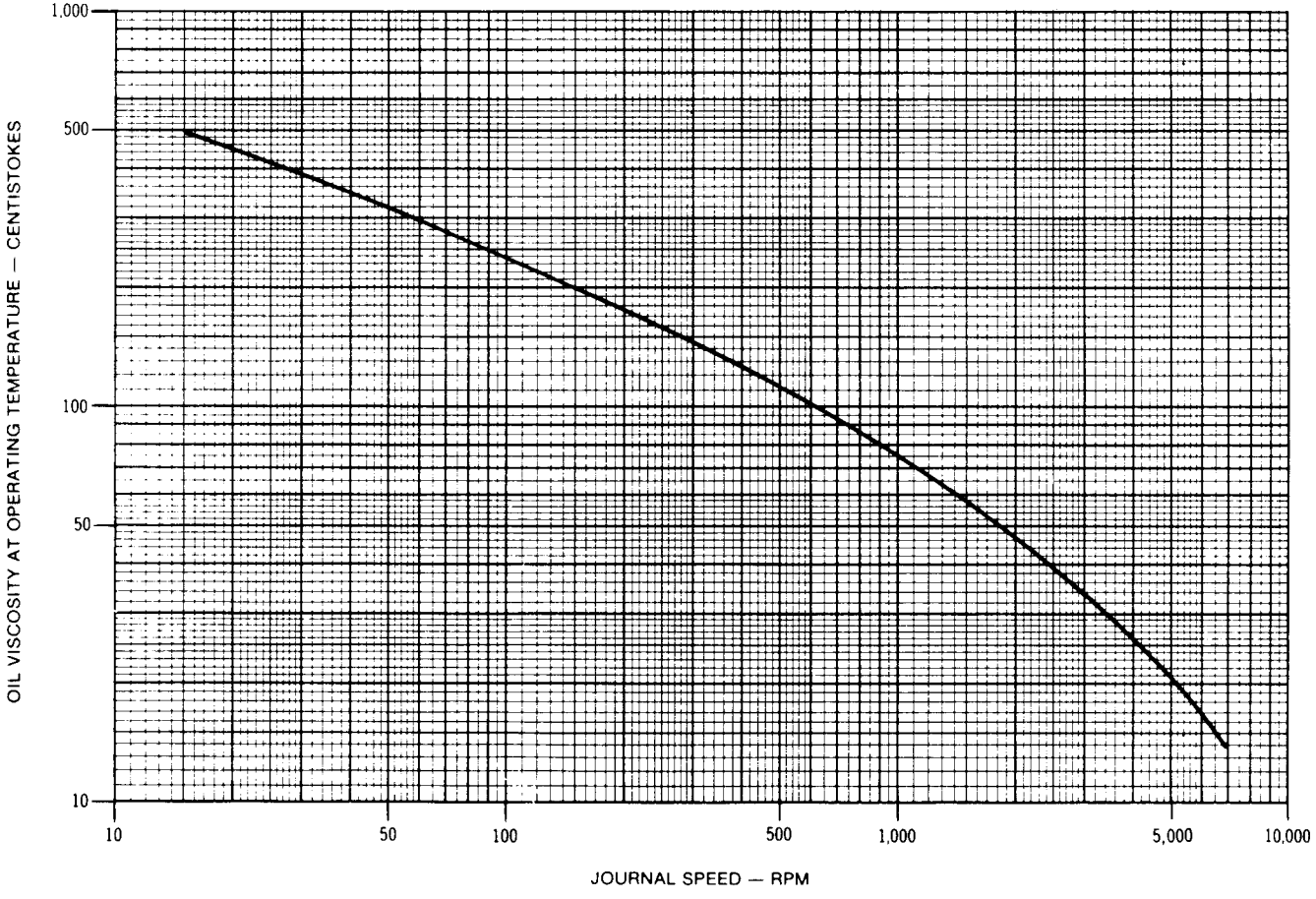

Figure 8.38 Oil viscosity chart for thin film bearings. Since loads can be assumed to be heavy enough for the bearing to operate in a thin film regime, only journal speed and operating temperature affect oil viscosity selection.

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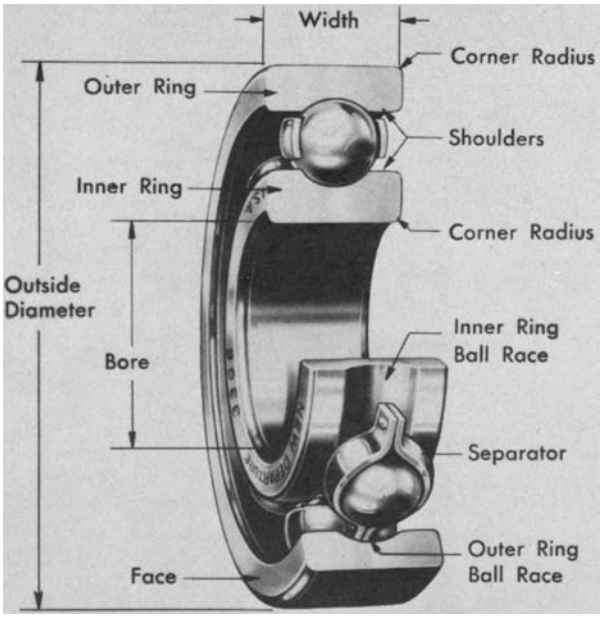

Figure 8.39 Ball bearing nomenclature.

raceways are machined in the rings to confine and guide the rolling elements. These rolling elements are usually held apart from each other, and their relative positions maintained to keep the shaft or spindle centered by the separator, which is also called a cage or retainer. In full-complement bearings, the rolling elements completely fill the space between the rings and no separator is used. Rolling element bearings have been manufactured in sizes ranging from smaller than a pinhead to 18 ft (6 m) or more in outside diameter.

Manufacturers now supply a wide variety of designs of rolling element bearings, of which those in Figures 8.40 and 8.41 represent only some of the more popular types. The bearing in Figure 8.39 is a single-row, deep-groove ball bearing, which is usually the initial choice for any application. One of the other types can be selected when the fatigue life of this design is inadequate, when space is limited, when self-alignment is required, when thrust loads must be carried, or when any of a variety of other conditions must be met. When one of these other types is selected, the desired performance characteristics are very often obtained at the expense of higher cost, a lower speed limit, or more severe lubrication requirements.

When a rolling element bearing is properly lubricated, its load capacity and life are limited primarily by the fatigue strength of the bearing steel. Normal rolling action applies repeated compressive loading in the contact area with stresses up to about 400,000 psi (2.8 GPa). In addition to causing elastic deformation of the rolling element and raceway, this stress induces shearing stress in the steel in a zone approximately 0.002–0.003 in. (0.05–0.075 mm) below the surface. This shearing stress induces fatigue cracks in the steel, which gradually grow and intersect. Small surface areas loosen and break away, forming pits. The actual time required for this to occur depends on many factors, including load, speed, and continuity of service, as well as the fatigue strength of the bearing steel.

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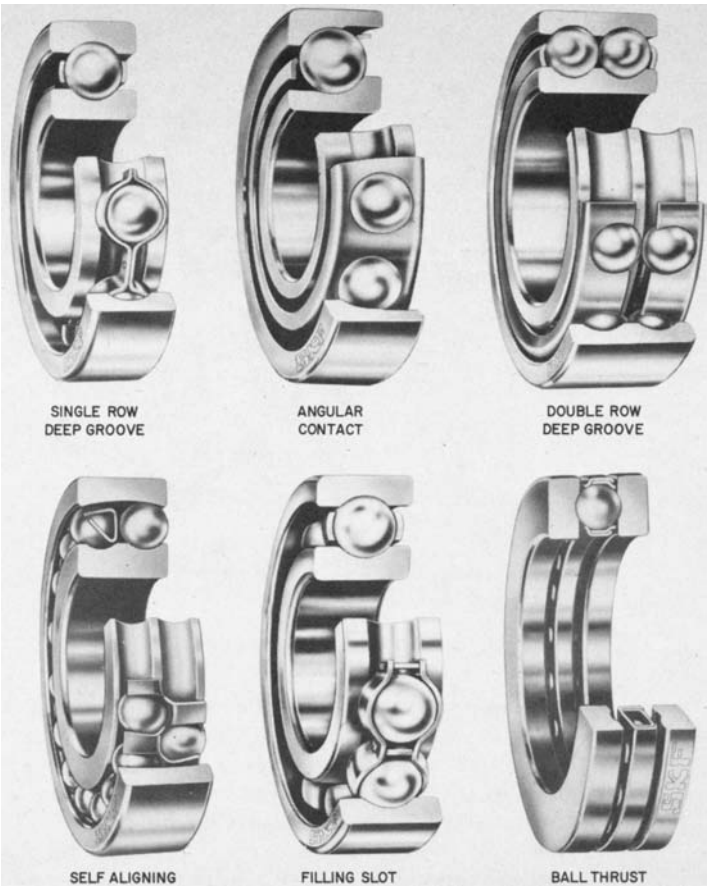

Figure 8.40 Popular types of ball bearing.

As discussed earlier in connection with elastohydrodynamic lubricating films (Section I.A.2), inadequate lubrication can greatly reduce the fatigue life of rolling element bearings. In addition, current research has shown that chemical effects have a considerable influence on fatigue life. The chemical composition of the lubricant additives, the base stock, and the contact surface materials are the major chemical variables. The water content of the atmosphere and the lubricant may also contribute significantly to chemical effects. At operating temperatures below 212°F (100°C), most industrial oils may contain dissolved water. Because the water molecules are small in comparison to oil molecules, they readily diffuse to the tips of the initial microcracks resulting from cyclic stressing. Although the precise mechanism by which this water accelerates fatigue cracking is not clear, there is evidence that the water in the microcracks breaks down and liberates atomic hydrogen, which attaches itself to the metal below the surface. This causes hydrogen embrittlement. Research data indicate that water-induced fatigue can reduce bearing life by 30–80%.

A. Need for Lubrication

Operation of rolling element bearings always results in the development of a certain amount of friction that comes from three main sources:

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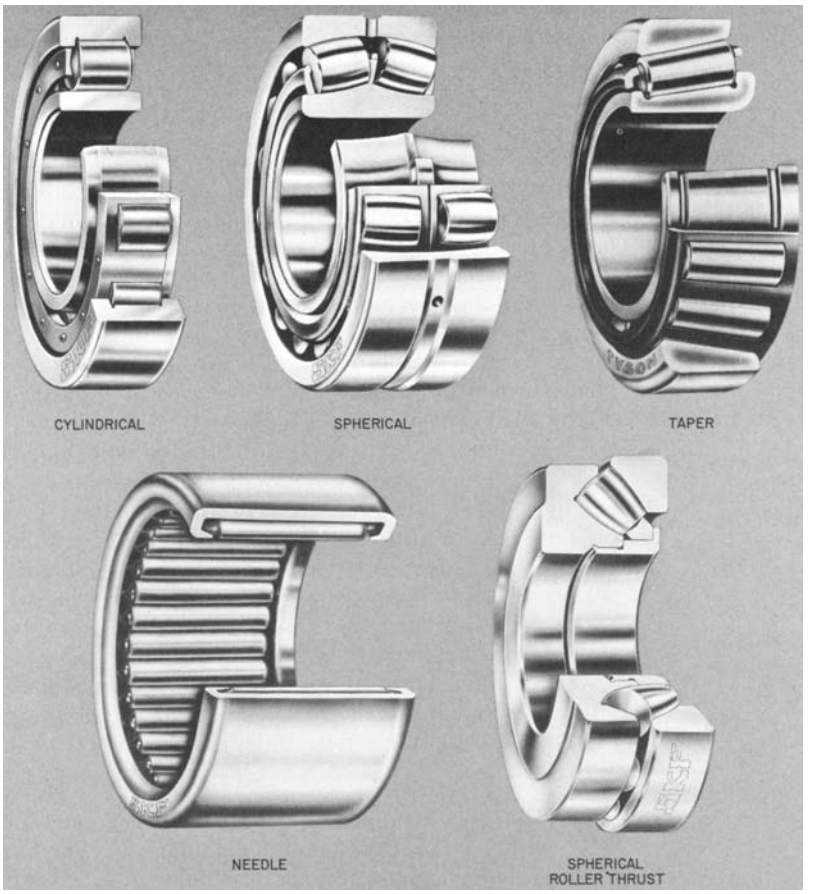

Figure 8.41 Popular types of roller bearing.

1. The fluid friction from shearing of the lubricating films
2. Displacement or churning of lubricant in the path of the rolling elements
3. Sliding or contact between various elements of the bearing

Sliding occurs in the following main areas:

1. Slipping between rolling elements and the raceways (sometimes referred to as skidding). Because of the elastic deformation of both the raceway and the rolling elements, true rolling does not occur and slight sliding results.
2. Sliding between the rolling elements and the separator (cage).
3. Sliding between the end of rollers and the raceway flanges of roller bearings. This type of sliding may be particularly severe in bearings of certain types designed for thrust loads.
4. Sliding between the shaft or spindle and contact-type housing seals.
5. Sliding between adjacent rolling elements in full-complement bearings.

Lubrication aims to maintain suitable films between all these sliding parts—EHL films for the contacts between the rolling elements and raceways, or thin films for other

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sliding parts. These films must be adequate to minimize friction and protect against wear. In addition, lubrication is expected to protect against rusting or other corrosive effects of contaminants and may provide part of the sealing against contaminants. In circulation systems, the lubricant also acts as a coolant to remove heat generated by friction, or heat conducted to the bearing from outside sources.

B. Factors Affecting Lubrication

Whether oil or grease is used for rolling element bearings depends on a number of factors. For bearings installed in machines that require oil for other elements, the same oil is often supplied to the bearings. In other cases, it may be more convenient to lubricate the bearings separately, and grease is often used. Groups of similar bearings may be lubricated with oil by a circulation system, an oil mist system, or with grease in a centralized lubrication system. Many bearings are now ''packed for life'' with grease by the bearing manufacturer and need no further lubrication in service. The characteristics of these various methods of application have some influence on the oil or grease selected.

The thickness of the elastohydrodynamic films formed in the contact areas between rolling elements and the raceways is a function of the speed at which the surfaces roll together, the load, the oil viscosity, and the operating temperature. The film thickness increases with increases in speed or oil viscosity and decreases with increases in load or operating temperature (since higher temperatures reduce the oil viscosity). For any given set of operating conditions, the oil viscosity (or the viscosity of the oil in a grease) should be selected to provide a safe minimum film thickness.

The lubrication requirements of the other sliding surfaces of a bearing must also be considered in the selection of a lubricant, but in general, the primary consideration in lubrication selection is the requirements of the EHL films.

1. Effect of Speed

The speed at which the surfaces of a rolling element bearing roll together is a fairly complex calculation, so an approximation called the bearing speed factor, nd_m, is usually used. The bearing speed factor is determined by multiplying the rotational speed in revolutions per minute n by the pitch diameter in millimeters d_m. For convenience, the pitch diameter is taken as half the sum of the bearing bore d and the bearing outside diameter D.* That is:

$$nd_m = \frac{n(d + D)}{2}$$

Manufacturers have established maximum bearing speed factors for both oil and grease lubricated bearings. These factors are shown in Table 8.1 for commercial grade bearings. Precision bearings are rated 5–50% higher. The higher speed factors allowed for oil lubrication generally reflect the better cooling available with oil, and the lower fluid friction of low viscosity oils.

* The older *DN* factor (bearing bore in millimeters, *D,* multiplied by the speed in rpm, *N*) is still found in some references, although it has not been officially used by the bearing manufacturers' associations for many years. It is not a satisfactory factor to use because for a given bearing bore there are bearings with considerable variations in outside diameter and size of the rolling elements. Thus, for any given bore diameter there can be considerable variation in the speeds with which the elements roll together.

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Table 8.1 Bearing Speed Factors nd_m

Bearing type	Oil lubricated[a]	Grease lubricated
Radial ball bearings	500,000	340,000
Cylindrical roller bearings	500,000	300,000
Spherical roller bearings	290,000	145,000[b]
Thrust–ball and roller bearings	280,000	140,000

[a] Oil lubrication is preferred where heat dissipation is required.
[b] Grease lubrication is not recommended for spherical roller thrust bearings.

For bearings lubricated with grease, the considerations relative to bearing speed are mainly concerned with the physical properties of the grease. At low to moderate speeds, the grease must be soft enough to slump slowly toward the rolling elements, but it must not be so soft that excess material gets into the path of the rolling elements. Excess grease increases shearing friction and can cause high operating temperatures. For high speeds, a relatively stiff grease is sometimes used, but it must not be so stiff that the rolling elements, after cutting a channel through the grease, are unable to pick up and distribute sufficient amounts of grease to provide continuous replenishment of the lubricating films. Moreover, to keep shearing friction low and to prevent leakage from housing seals, the grease must have good resistance to softening due to mechanical shearing.

2. Effect of Load

Under EHL conditions, the effect of load on film thickness is not as great as the effect of speed or oil viscosity. For example, while doubling the speed or oil viscosity might result in a film thickness increase exceeding 50%, doubling the load might result in only about a 10% decrease in film thickness. As a result, under steady load conditions the viscosity of oil can generally be selected on the basis of the bearing speed factor and operating temperature without regard to the load.

When shock or vibratory loads are present, somewhat greater film thickness is necessary to prevent metal-to-metal contacts through the film; thus, higher viscosity oils are usually required. Under severe shock loading conditions, lubricants with enhanced antiwear properties may be desirable.

Pressures between the rolling elements and separators (cages) are not high enough to form EHL films. Additionally, these pressures act more or less continuously on the same spots in the separator pockets. As a result, thin film lubrication exists, and sometimes a higher viscosity oil not only improve the resistance to wiping of the oil films but can reduce wear of the separator, as well. When grease is the lubricant, wear can be reduced only if the lubricant can penetrate the clearances between the rolling elements and separators. This is especially important with machined separators where clearances are small.

In tapered and spherical roller thrust bearings, heavy loads sometimes force the ends of the rollers against the flanges of the raceways with considerable pressure. Severe wiping of the lubricant films and wear of the rollers and flanges may occur. In bearings of these types, a lubricant with enhanced antiwear properties may be required, even though the rollers and raceways can be lubricated satisfactorily with conventional products.

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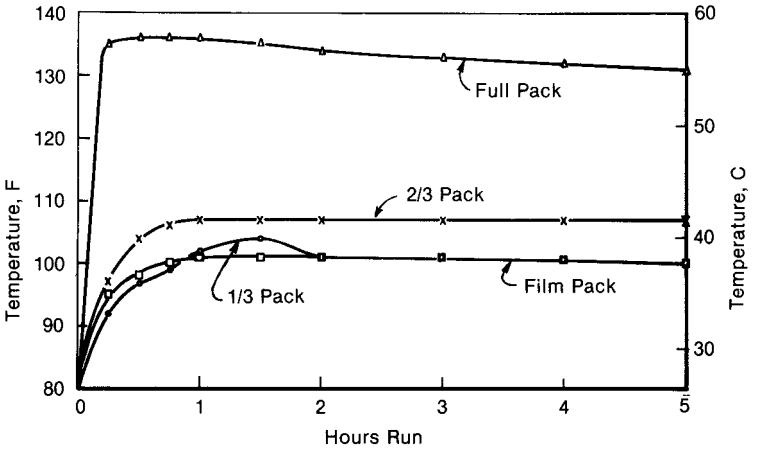

Figure 8.42 Temperature rise in grease-lubricated bearings. As shown by the curves, the final operating temperature of a bearing is a function of the amount of grease packed in it.

3. Effect of Temperature

Since both viscosity and grease consistency are functions of temperature, the operating temperature of bearings must always be considered during selection of lubricants. Bearing operating temperatures may be increased above normal by heat that is conducted to the bearing from a hot shaft or spindle, or by heat radiated to the housing from a hot surrounding atmosphere. Excessive churning of grease resulting from overfilling can also raise bearing temperatures (Figure 8.42). Higher than normal temperatures reduce oil viscosity, whereupon film thickness may decrease below a safe level, and grease may soften so much that excessive churning and further frictional heating occur.

High temperatures also increase the rate at which both oil and grease deteriorate due to oxidation. Oxidation may result in thickening of oil and eventually could lead to deposits that will interfere with oil flow or bearing operation. Oxidation may also lead to deposits with greases, and in severe cases, to hardening such that the grease cannot feed and lubricate.

Under low temperature conditions, the lubricant must be such that the bearing can be started with the power available, and it must distribute sufficiently to prevent excessive wear before frictional heating warms the bearing and lubricant.

4. Contamination

Solid particles of any kind that are trapped between the rolling elements and raceways are the most frequent cause of shortened bearing life. Consequently, dirt should be kept out of bearings as much as possible (including during periods of storage prior to installation), and lubricants should be changed before oxidation has progressed to a point at which deposits begin to form. The use of premium quality, oxidation-inhibited lubricants can greatly extend the period of time that lubricants may be left in service without excessive oxidation.

Water that gets into a bearing tends to reduce the fatigue life and cause rusting that can quickly ruin a bearing. Water, when mixed with some greases, may cause them to

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soften and perhaps leak from the bearings. Large quantities of water can wash the lubricant out. Sometimes fluids such as acids get into the bearings and cause corrosion. Any of these conditions usually involves special precautions and possibly specialty lubricants.

C. Lubricant Selection

Rolling element bearings generally require high quality lubricants, many of which are specially formulated for the purpose.

1. Oil Selection

Oils for the lubrication of rolling element bearings should have the following characteristics:

1. Excellent resistance to oxidation at operating temperatures, to provide long service without thickening or formation of deposits that would interfere with bearing operation.
2. Proper viscosity at operating speeds and temperatures to protect against friction and wear. Two charts (Figure 8.43) may be used to select the proper viscosity at the operating temperature from the type of bearing, the operating speed, and bearing outside diameter (D). The speed factor (SF) is determined for the speed and type of bearing from Figure 8.43a. The bearing size/speed factor ($BS/SF = SF \times D$) is used in Figure 8.43b to determine the correct oil viscosity for a given specific film thickness (λ). For steady loads, the minimum viscosity line, corresponding to a specific film thickness, $\lambda = 1.5$ (see discussion of EHL in Section I.A.2), will provide adequate film thickness. When shock or vibratory loads are present, higher viscosity and thicker films are desirable. Therefore, the optimum viscosity line, corresponding to a specific film thickness, $\lambda = 4.0$, should be used. ASTM D 341, Standard Viscosity–Temperature Charts for Liquid Petroleum Products, should be used to translate this viscosity to the viscosity at the standard reporting temperatures. EHL calculations should be used for more accurate determination of viscosity requirements.
3. Antirust properties to protect against rust when moisture is present.
4. Good antiwear properties where required because of heavy or shock loads, or where thrust loads cause heavy wiping between the ends of rollers and raceway flanges.
5. Good demulsibility to allow separation of water in circulation systems.

In addition to these characteristics, increasing emphasis is now being placed on the interrelationship between oil characteristics and fatigue. The specific lubricant properties that influence fatigue are not easily defined; but where average bearing life is lower than expected, special lubricants formulated to help minimize fatigue may be desirable. The combination of select base stocks and specific additives can enhance the antifatigue properties of lubricants.

2. Grease Selection

Grease for the lubrication of rolling element bearings should have the following characteristics:

Excellent resistance to oxidation to resist the formation of deposits or hardening that might shorten bearing life. Greases used in ''packed for life'' bearings are expected to provide trouble-free performance for the life of the bearing, assuming normal operating temperatures.

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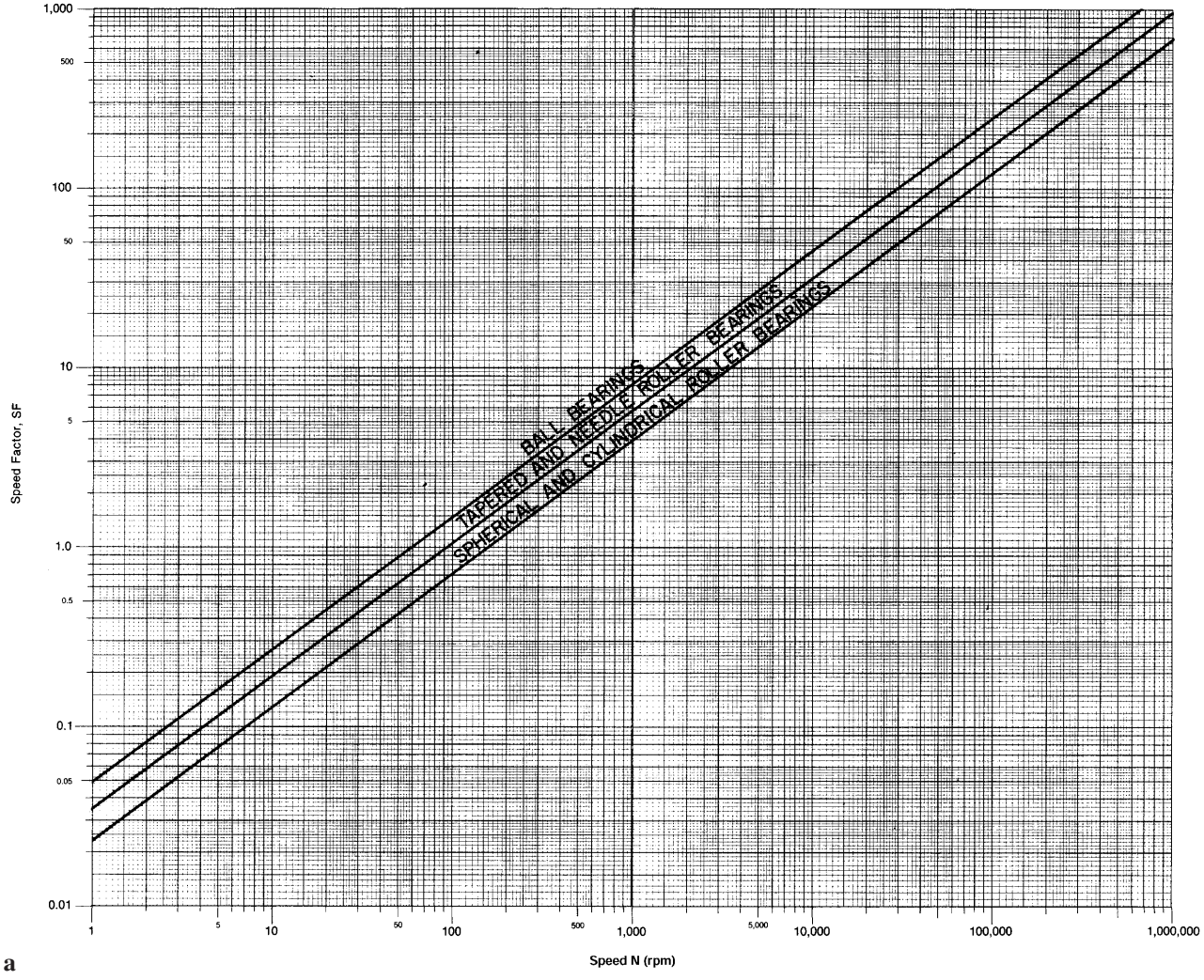

a

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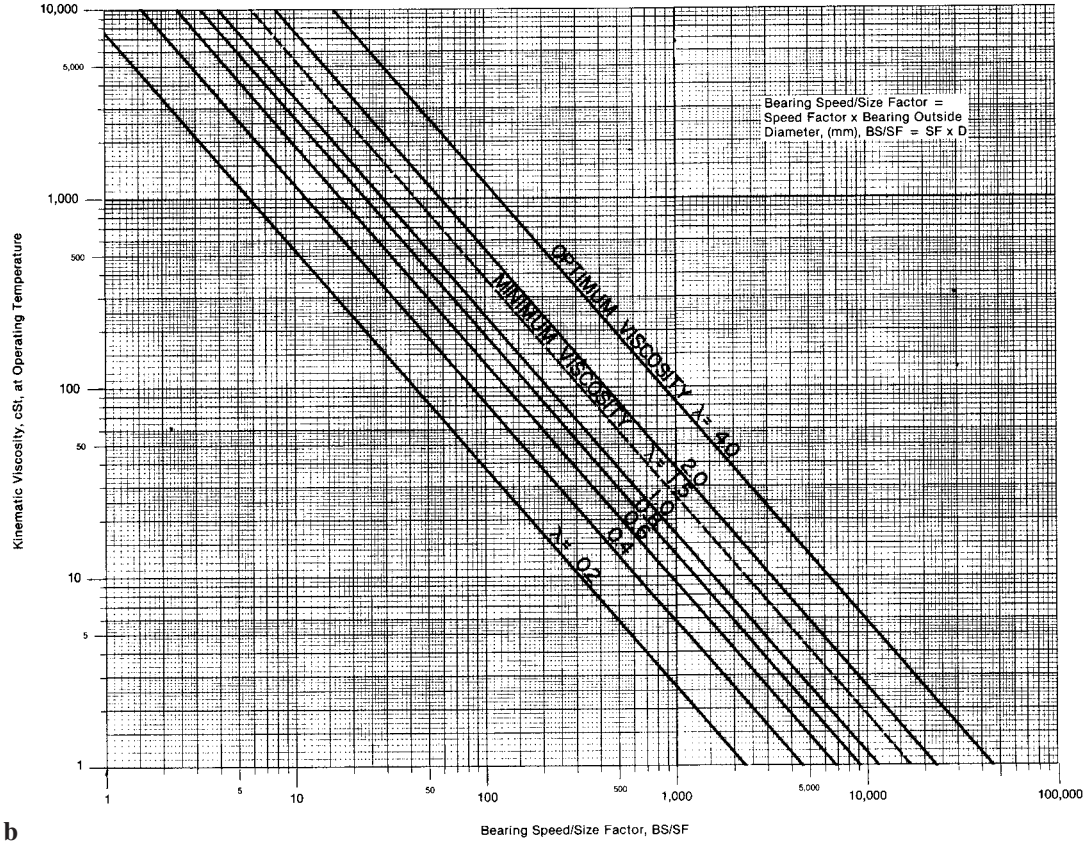

b

Figure 8.43 Viscosity selection charts for rolling element bearings. (a) For normal loads oil viscosity selected with the minimum viscosity line is generally satisfactory. (b) For shock or vibratory loads, somewhat thicker oil films are required and the optimum viscosity line should be used.

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Mechanical stability to resist excessive softening or hardening as a result of shearing in service.

Proper consistency for the method of application and proper feedability at operating temperatures to maintain adequate lubricating films without excessive slumping, which would increase friction and bearing heating.

Controlled oil bleeding, particularly for high speed bearings, to supply the small amount of oil needed to form EHL films.

Enhanced antiwear properties to resist the wiping action between roller ends and raceway flanges in bearings carrying heavy radial or axial thrust loads.

Ability to protect surfaces against rusting. When small quantities of water contaminate a bearing, the grease should absorb the water without appreciable softening or hardening. When large quantities of water are present, the grease must resist being washed out.

Good compatibility with system and system components.

Depending on application, the ability to resist corrosion or deterioration where acids or caustic materials in small quantities can get into the grease.

IV. SLIDES, GUIDES, AND WAYS

The category of bearings sometimes called flat bearings includes all slides, guides, and ways used (1) on forging and stamping presses, (2) as crosshead guides of certain compressors, diesel and steam engines, and (3) on metalworking machines such as lathes, grinders, planers, shapers, and milling machines. The service conditions under which these bearings operate vary widely. Crosshead guides operate at relatively high speeds under conditions that permit the formation of fluid lubricating films most of the time. The requirements for lubrication usually are not severe, and the guides usually are designed to operate on the same oil used for the main and connecting rod bearings of the machine.

The lubrication of ways and slides of machine tools can present special problems. At low speeds and under heavy loads, the lubricant tends to be wiped off causing boundary lubrication to prevail. While this results in higher friction, boundary films have the advantage of being almost constant in thickness. With low loads and high traverse speeds, the oil viscosity must be high enough to allow the formation of fluid films that will lift and float the slide. With variations in speed or load, these fluid films can vary in thickness enough to produce wavy surfaces on the parts being machined, or cause them to run offsize. Thus, precision machining generally requires that the slides and ways operate under boundary conditions at all times. This frequently means that friction-reducing and antiwear additives must be included in the oil.

The phenomenon known as stick–slip can be encountered in the motion of slides and ways. If the static coefficient of friction of the lubricant is greater than the dynamic coefficient, more force is required to start the slide from rest than is required to maintain it in motion after it has started. There is always some amount of free play in the feed mechanism; so when force is applied to start the slide in motion, there will be initial resistance. When the force becomes high enough, the slide will begin to move. As soon as motion begins, the force required to maintain motion decreases so the slide will jump ahead until the free play in the feed mechanism is taken up. At low traverse speeds, this can be a continuous process, producing chatter marks on the workpiece. With cross-slides, stick–slip effects can make it extremely difficult to set feed depths accurately. Stick–slip

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effects can be overcome with additives that reduce the static coefficient of friction to a value equal to or less than the dynamic coefficient.

With vertical guides, the lubricant tends to drain from the surfaces. To resist this tendency and secure adequate films, special adhesive characteristics are needed.

A. Film Formation

When the two parallel surfaces move with respect to each other, there is some tendency to draw oil in and form a fluid film between the surfaces. However, such a film does not have high load-carrying capacity, and when the surfaces are long, the film is usually wiped or squeezed away rapidly. Traverse grooves in the slide divide it into a series of shorter surfaces, where lubricant supplied at the grooves can improve film formation and load-carrying ability. In some cases, with precision machine tools, hydrostatic bearings are being used. Pockets are machined in the slide and supplied from a constant pressure system with a flow restrictor for each pocket. This arrangement eliminates stick–slip effects and, if properly arranged, maintains the slide parallel to the way. More and more hydrostatic systems are being used in industry, particularly where stick–slip characteristics can contribute to reduced precision of machined parts.

1. Grease Lubrication

On some machine tools, grease is used for the lubrication of the slides and ways. Relative to oil, grease lubrication provides some advantages and disadvantages that should be recognized. The advantages are better sealing, improved stay-put properties, and less generally susceptibility to wash-off by water or metalworking fluids. The disadvantages are apparent when a high degree of accuracy in machined parts is necessary, and in applications in which the grease can act as a binder for debris and pose potential compatibility issues with metalworking fluids.

The potential disadvantage in accuracy of machined parts would be found in applications calling for tables or ways to operate with a specific film thickness at all times. Greases have a greater tendency to build thicker films when initially applied, and these films are reduced as the grease is squeezed out or temperatures rise. Greases can also act as a binder for chips, grinding compounds, and other materials present in machining operations. These could damage slides and ways if pulled into the contact areas. Excessive grease that could contaminate coolants or metalworking fluids may shorten batch life or reduce performance.

2. Lubricant Characteristics

On the basis of the foregoing requirements, the characteristics of suitable lubricants for slides, guides, and ways can be summarized as follows:

- Proper viscosity at operating temperature for ready distribution to the sliding surfaces and for forming the necessary boundary films
- High film strength to maintain the required boundary films under heavy loads and antiwear capability to control wear under these boundary conditions
- Proper frictional characteristics to prevent stick–slip and chatter
- Adequate adhesiveness to maintain films on intermittently lubricated surfaces, especially vertical surfaces, and to resist the washing by any metalworking coolants used

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Slides and guides in some machines such as open-crankcase steam engines and forging machines may be exposed to considerable amounts of water. Lubricants for these applications must be specially formulated to resist being washed off.

V. GEARS

Gears are employed to transmit motion and power from one rotating shaft to another, or from a rotating shaft to a reciprocating element. With respect to lubrication and the formation and maintenance of lubricating films, gears can be classified as follows:

Spur (Figure 8.44), bevel (Figure 8.45), helical (Figure 8.46), herringbone (Figure 8.47), and spiral bevel (Figure 8.48) gears
Worm gears (Figure 8.49)
Hypoid gears (Figure 8.50)

The differences in the action between the teeth of these three classes of gears have a considerable influence on both the formation of lubricating films and the properties of the lubricants required for satisfactory lubrication.

A. Action Between Gear Teeth

As gear teeth mesh, they roll and slide together. The progression of contact as a pair of spur gear teeth of usual design engage as shown in Figure 8.51. The first contact is between a point near the root of the driving tooth (upper gear) and a point at the tip of the driven tooth. In view A, these points are identified as 0–0 lying on the line of action. At this

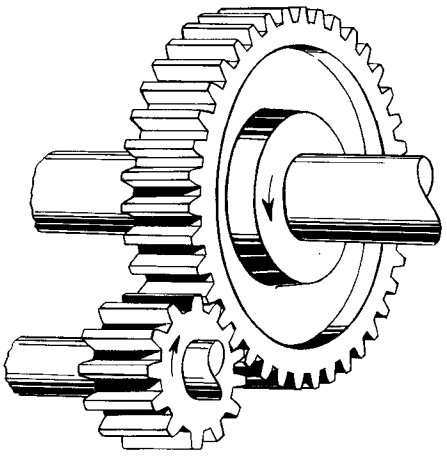

Figure 8.44 Spur gears.

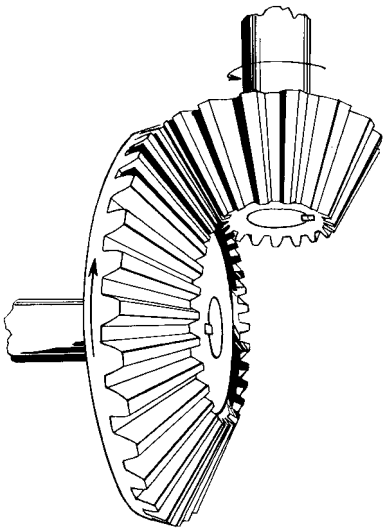

Figure 8.45 Bevel gears. These gears transmit motion between intersecting shafts. The shafts need not meet at a right angle.

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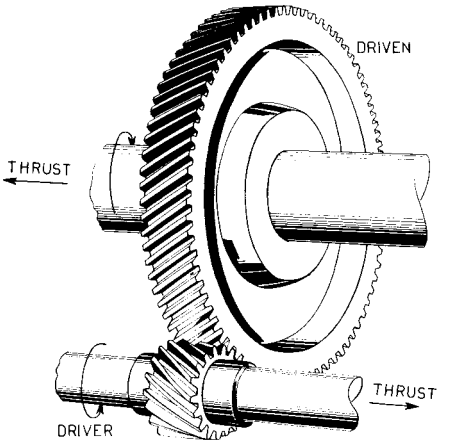

Figure 8.46 Helical gear and pinion.

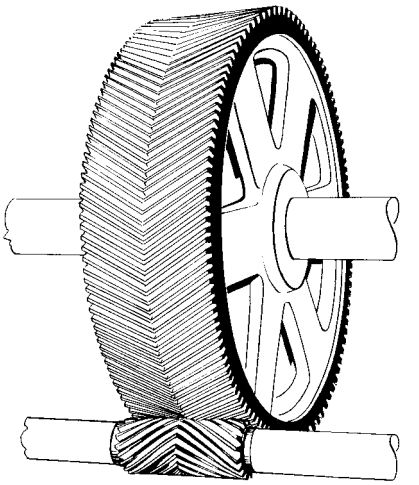

Figure 8.47 Herringbone gear and pinion.

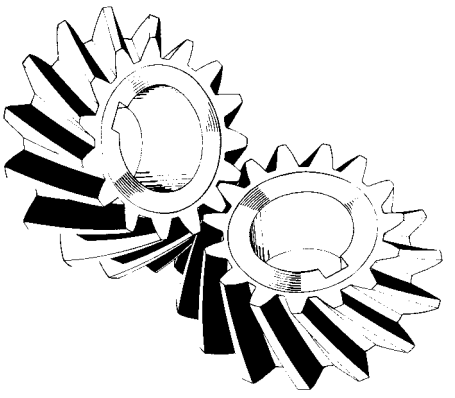

Figure 8.48 Spiral bevel gears. These gears transmit motion between intersecting shafts.

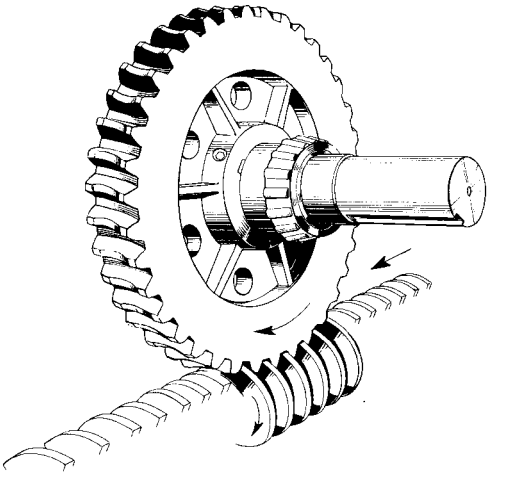

Figure 8.49 Worm gears; the worm here is represented by an endless rack.

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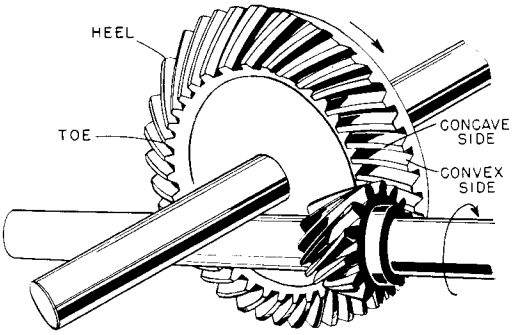

Figure 8.50 Hypoid gears. These gears transmit motion between nonintersecting shafts crossing at a right angle.

time, the preceding teeth are still in mesh and carrying most of the load. As contact progresses, the teeth roll and slide on each other. Rolling is from root to tip on the driver and from tip to root on the driven tooth. The direction of sliding at each stage of contact is as indicated by the small arrows.

In view B, contact has advanced to position 3–3, which is approximately the beginning of ''single tooth'' contact when one pair of teeth pick up the entire the load. It will be seen that to reach this point of engagement, since the distance 0–3 on the driven gear is greater than the distance 0–3 on the driver, there must have been sliding between the two surfaces. View C, position 4–4, shows contact at the pitch line, where there is pure rolling—no sliding. It should be noted, particularly, that the direction of sliding reverses at the pitch line. Also, sliding is always away from the pitch line on the driving teeth, and always toward it on the driven teeth. View D shows contact at position 5–5, which marks the approximate end of a single-tooth contact. As shown, another pair of teeth is about to make contact. In view E, two pairs of teeth are in mesh, but shown at position 8–8, the original pair of teeth is about to disengage.

It will be seen that rolling is continuous throughout mesh. Sliding, on the other hand, varies from a maximum velocity in one direction at the start of mesh, through zero velocity at the pitch line, then again to a maximum velocity in the opposite direction at the end of mesh.

This combination of sliding and rolling occurs with all meshing gear teeth regardless of type. The two factors that vary are the amount of sliding in proportion to the amount of rolling, and the direction of slide relative to the lines of contact between tooth surfaces.

With conventional spur and bevel gears, the theoretical lines of contact run straight across the tooth faces (Figure 8.52). The direction of sliding is then at right angles to the lines of contact. With helical, herringbone, and spiral bevel gears, because of the twisted shape of the teeth, the theoretical lines of contact slant across the tooth faces (Figure 8.53). Therefore, the direction of sliding is not at right angles to the lines of contact, and some side sliding along the lines of contact occurs.

With worm gears, as with spur gears, the same sliding and rolling action occurs as the teeth pass through mesh. Usually, this sliding and rolling action is relatively slow because of the low rotational speed of the worm wheel. In addition, rotation of the worm

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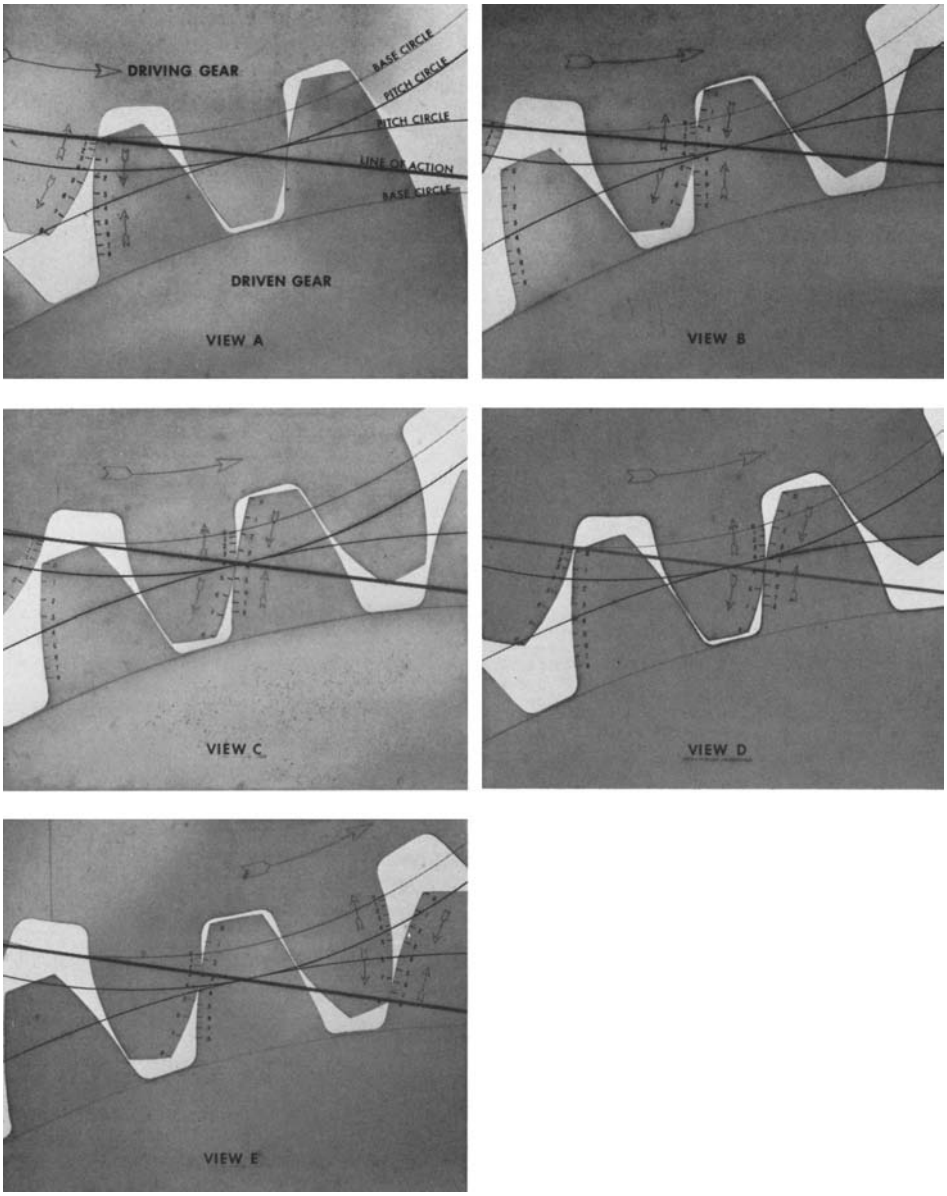

Figure 8.51 Meshing of involute gear teeth: the progression of rolling and sliding as a pair of involute gear teeth (a commonly used design) pass through mesh. The amount of sliding can be seen from the relative positions of the numbered marks on the teeth.

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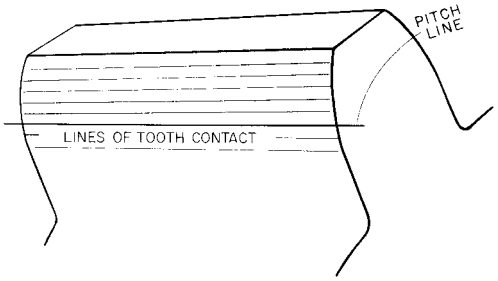

Figure 8.52 A spur gear tooth showing lines of tooth contact. On the driving tooth, this contact first occurs below the pitch line. As the gear turns, this contact progressively sweeps upward to the top of the tooth. The action is reversed on the driven tooth.

introduces a high rate of side sliding. The combination of two sliding actions produces a resultant slide, which in some areas is directly along the line of contact.

In addition to the usual rolling action, hypoid gears have a combination of radial and sideways sliding that is intermediate between the motions of worm gears and spiral bevel gears. The greater the shaft offset, the more nearly the sliding conditions approach those found in worm gears.

B. Film Formation

As pointed out earlier, for EHL film formation to occur, the pressure must be high enough to cause elastic deformation of the contacting surfaces. Thus, the theoretical lines or points of contact are expanded into areas, and a convergent zone exists ahead of the contact area. In the case of industrial gears, in all except very lightly loaded applications, tooth loading is high enough to produce elastic deformation along the line of contact. A convergent zone immediately ahead of the contact area also exists at all times (see Figure 8.54), and the conditions for the formation of thick EHL films are complex, and are beyond the scope

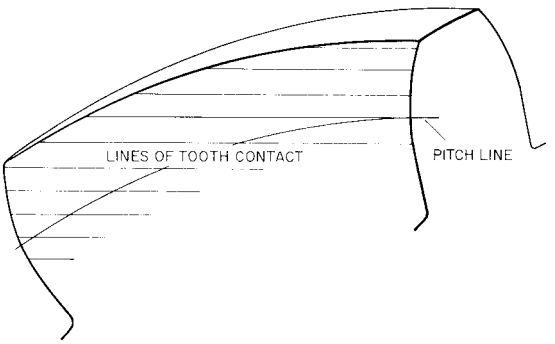

Figure 8.53 Tooth contact of low angle helical gear.

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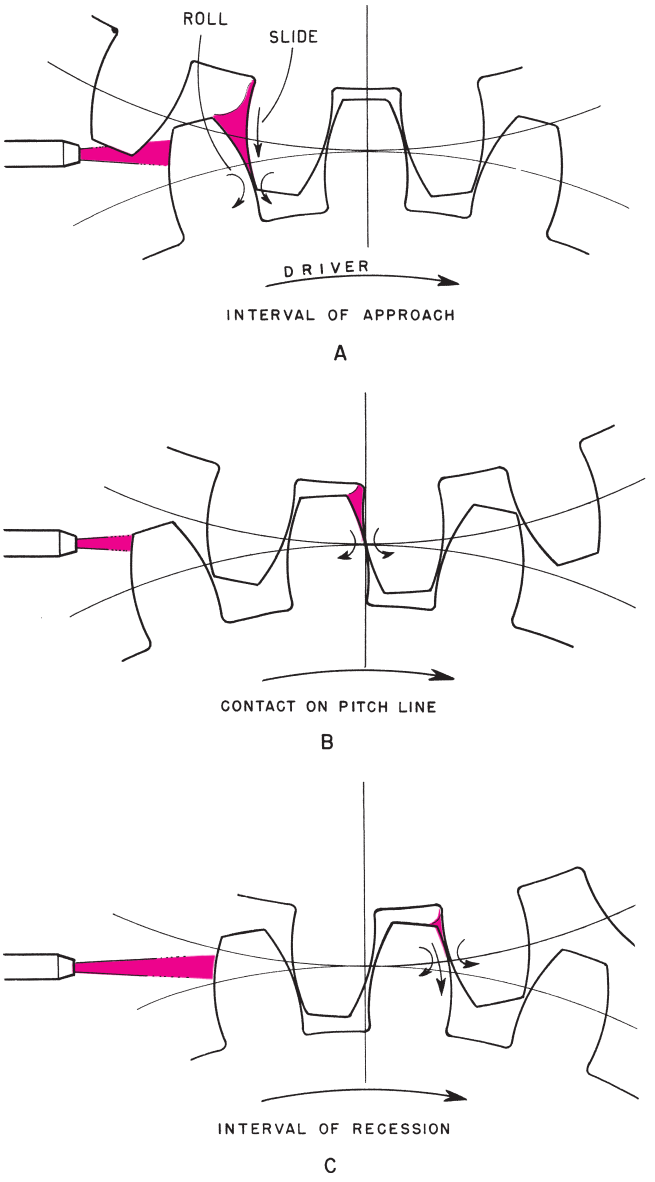

Figure 8.54 Convergent zone between meshing gear teeth. Clearly, if oil is present between meshing gear teeth, it will be drawn into the convergent zone between the teeth; the point of this wedge-shaped zone always points toward the roots of the driving teeth.

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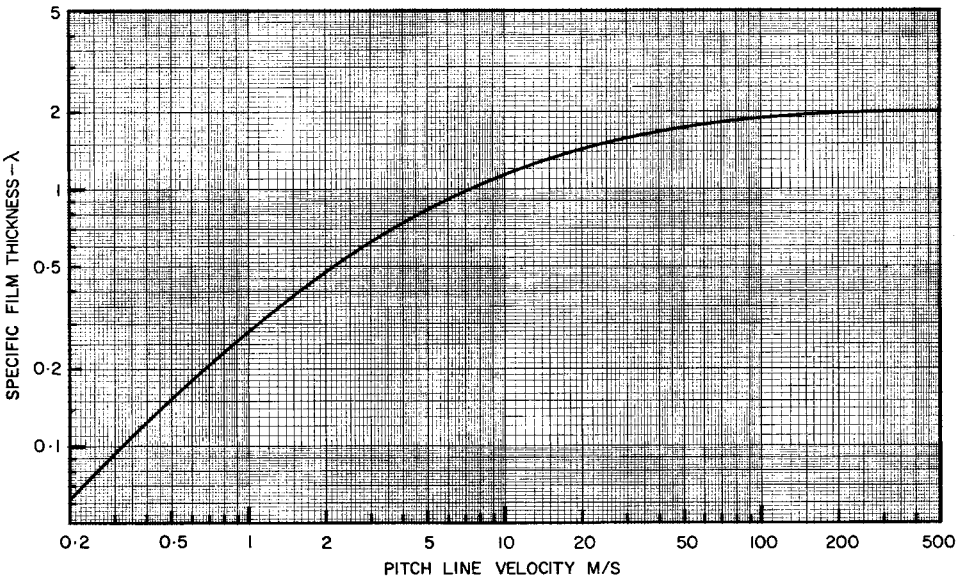

Figure 8.55 Critical specific film thickness for gears: the curve is based on a 5% probability of surface distress to define the target film thickness, which is adjusted to reflect the root mean square (rms) surface roughness, $\sigma = (\sigma_2^2 + \sigma_2^2)^{1/2}$.

of this publication.* However, certain factors in these calculations are of importance in the general consideration of selection of lubricants for industrial gear drives. The equations used do not consider the effect of tooth sliding action on the formation of the EHL films. The entraining velocity tending to carry the lubricant into the contact zone is considered to be the rolling velocity alone. The rolling velocity, for convenience, is usually calculated at the pitch line and is taken to be representative for the entire tooth.

The critical specific film thickness λ for gears is not only considerably lower than for rolling element bearings but is also a function of the pitch line velocity. The curve of Figure 8.55, developed from experimental data, shows that at low speed values of λ of 0.1 or lower can be tolerated without surface distress in the form of pitting or wear. At higher speeds, values of λ of up to 2.0 or higher may be required for equal freedom from tooth distress.

Currently, no analysis has been made of the reasons for these lower specific film thicknesses providing satisfactory results in gears. However, it is generally accepted that in the range where $\lambda < 1.0$, lubricants containing extreme pressure and antifatigue additives are required.

In the selection of lubricants for gears, tooth sliding is considered from two aspects:

1. It tends to increase the operating temperature because of frictional effects.
2. Sliding along the line of contact tends to wipe the lubricant away from the convergent zone; thus, it is more difficult to form lubricating films.

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* An excellent reference on this subject is the Mobil EHL Guidebook.

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C. Factors Affecting Lubrication of Enclosed Gears

The lubricant in an enclosed gear set, which represents the major portion of gear usage, is subjected to very severe service, being thrown from the gear teeth and shafts in the form of a mist or spray. In this atomized condition, it is exposed to the oxidizing effect of air. Fluid friction and, in some cases, metallic friction generate heat, which raises the lubricant temperature. The violent churning and agitation of the lubricant by the gears of splash-lubricated sets also raises the temperature. Raising the temperature increases the rate of oxidation. Sludge or deposits, formed as a result of oil oxidation, can restrict oil flow, or interfere with heat flow in oil coolers or heat dissipation from the sides of the gear case. Restrictions in the oil flow may cause lubrication failure, while heat-insulating deposits decrease cooling and cause further increases in the rate of oxidation. Eventually, lubrication failure and damage to the gears may result.

In selecting the lubricant for enclosed gear sets, in addition to the requirement for adequate oxidation resistance, the following factors of design and operation require consideration:

1. Gear type
2. Gear speed
3. Reduction ratio
4. Operating and start-up temperatures
5. Transmitted power
6. Surface finish
7. Load characteristics
8. Drive type
9. Application method
10. Contamination (water, metalworking fluids, dirt, etc.)
11. Lubricant leakage

1. Gear Type

With spur and bevel gears, the line of contact runs straight across the tooth face, and the direction of sliding is at right angles to the line of contact. Both these conditions contribute to the formation of effective lubricating films. However, only a single tooth carries the whole load during part of the meshing cycle, resulting in high tooth loads. Additionally, if one tooth wears, there is no transfer of load to other meshing teeth to relieve the load on the worn tooth, and wear of that tooth will continue.

Helical, herringbone, and spiral bevel gears always have more than one pair of teeth in mesh. This results in better distribution of the load under normal loading. Under higher loading, the individual tooth contact pressures may be as high as in comparable straight tooth gears under normal loading. The sliding component along the line of contact, because of time and high viscosity of the lubricant in the contact area, has little or no effect on the EHL film in the contact area. In the convergent zone ahead of the contact area, the sliding component tends to wipe the lubricant sideways. Therefore, not as much lubricant is available to be drawn into the contact area, and the resultant pressure increase in the convergent zone may not be as great. These effects may contribute to a need for slightly higher viscosity lubricants, although, in general, oils for gears of these types are selected on the same basis as for straight tooth gears.

An additional factor present with helical, herringbone, and spiral bevel gears is that if one tooth wears, the load is transferred simultaneously to other teeth in mesh. This

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relieves the load on the worn tooth and may make lubricant characteristics somewhat less critical for gears of these types. With all such gears, it is important that the lubricant have a viscosity high enough to provide effective oil films, but not so high that excess fluid friction will occur.

The high rate of side sliding in worm gears results in considerable frictional heating. Generally, the rolling velocity is quite low, so the velocity tending to carry the lubricant into the contact area is low. Combined with the sliding action tending to wipe the lubricant along the convergent zone, this makes it necessary to use high viscosity lubricants (typically ISO 460 or 680 viscosity grade). EP additive-type gear oils are not normally recommended for worm gears but, to help reduce the wiping effect and reduce friction, lubricants containing friction-reducing materials are usually used. Because of their friction-reducing and long life characteristics, synthetic lubricants (such as synthesized hydrocarbon or polyalkylene glycols) are the lubricant of choice for most worm gear applications.

Hypoid gears are of steel-to-steel construction and are heat-treated. They are designed to transmit high power in proportion to their size. Combined with the side sliding that occurs, these gears operate under boundary or mixed film conditions essentially all the time and require lubricants containing active extreme pressure additives.

2. Gear Speed

The higher the speed of meshing gears, the higher will be the sliding and rolling speeds of individual teeth. When an ample supply of lubricant is available, speed assists in forming and maintaining fluid films. At high speed, more oil is drawn into the convergent zone; in addition, the time available for the oil to be squeezed from the contact area is less. Therefore, comparatively low viscosity oils may be used (despite their fluidity there is insufficient time to squeeze out the oil film). At low gear speeds, however, more time is available for oil to be squeezed from the contact area and less oil is drawn into the convergent zone; thus, higher viscosity oils are required.

3. Reduction Ratio

Gear reduction ratio influences the selection of the lubricating oil because high ratios require more than one step of reduction. When the reduction is above about 3:1 or 4:1, multiple reduction gear sets are usually used and above about 8:1 or 10:1, they are nearly always used. In a multiple reduction set, the first reduction operates at the highest speed and so requires the lowest viscosity oil. Subsequent reductions operate at lower speeds so require higher viscosity oils. The low speed gear in a gear set is usually the most critical in the formation of an EHL film. In the case of a gear reducer, this would be the output gear. In very high speed gear reducers, both the lowest speed and highest speed gears should be checked to determine the more critical condition. In some cases, a dual viscosity system may be employed, using a lower viscosity oil for the high speed gears and a higher viscosity oil for the low speed gears. In some gear sets, this can be accomplished automatically by circulating the cool oil first to the low speed gears, and then, after it is heated and its viscosity decreased, to the high speed gears.

4. Operating and Start-Up Temperatures

The temperature at which gears operate is an important factor in the selection of the lubricating oil, since viscosity decreases with increasing temperature and oil oxidizes more rapidly at high temperatures. Both the ambient temperature where the gear set is located and the temperature rise in the oil during operation must be considered.

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When gear sets are located in exposed locations, the oil must provide lubrication at the lowest expected starting temperature. In splash-lubricated units, this means that the oil must not channel at this temperature, while in pressure-fed gear sets, the oil must be fluid enough to flow to the pump suction. At the same time, the oil must have a high enough viscosity to provide proper lubrication when the gears are at their highest operating temperature. For gears exposed to low temperatures ($<0°C$) during start-up or in continuous operation, synthetic lubricants such as synthesized hydrocarbon oils (SHF) are most often recommended due to their very low pour point, high viscosity index, and excellent shear stability.

During operation, the heat generated by metallic friction, between the tooth surfaces and by fluid friction in the oil, will cause the temperature of the oil to rise. The final operating temperature is a function of both this temperature rise in the oil and the ambient temperature surrounding the gear case. Thus, a temperature rise of 90°F (32.2°C) and an ambient temperature of 60°F (15.6°C) will produce an operating temperature of 150°F (65.6°C), while the same temperature rise at an ambient temperature of 100°F (37.8°C) will produce an operating temperature of 190°F (87.8°C). In the latter case, an oil of higher viscosity and better oxidation stability would be required to provide satisfactory lubrication and oil life at the operating temperature. For gear sets equipped with heat exchangers in the oil system, both the ambient temperature and the temperature rise are less important, since the operating temperature of the oil can be adjusted by varying the amount of heating or cooling.

5. Transmitted Power

As noted in the discussion of EHL film formation, load does not have a major influence on the thickness of EHL films. However, it cannot be ignored. As load is increased, the viscosity of the lubricant may have to be increased to adjust for the small affect of load on film thickness, particularly where λ values were marginal prior to increasing load.

Load also has an influence on the amount of heat generated by both fluid and mechanical friction. Gears designed for higher power ratings will have wider teeth, teeth of larger cross section, or both. Regardless, a greater surface area is swept as the teeth pass through mesh, causing mechanical and fluid friction to be greater. At the same time, the relative area of radiating surface in proportion to the heat generated is usually less in a large gear set than it is in a small one. As a result, larger gear sets, transmitting more power, tend to run hotter unless they are equipped with oil coolers. However, if the operating temperature of a gear set is properly taken into account in the selection of lubricant viscosity, the heating effects based on the amount of power transmitted will be taken care of.

6. Surface Finish

As discussed, surface roughness has an important influence on the thickness of oil films required for proper lubrication. Rougher surfaces require thicker oil films to obtain complete separation, and higher viscosity oils. On the other hand, smoother surfaces can be lubricated successfully with lower viscosity oils. Since some smoothing of the surfaces results from running in, some authorities recommend using an estimated ''run in'' surface roughness rather than the ''as finished'' values in oil film thickness calculations and for selection of oil viscosity.

7. Load Characteristics

The nature of the load on any gear set has an important influence on the selection of a lubricating oil. If the load is uniform, the torque (turning effort) and the load carried by

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the teeth will also be uniform. However, excessive tooth loads due to shock loads may tend to momentarily rupture the lubricating films. Therefore, where the shock factor has not been considered in the design or selection of a gear set, a higher than normal oil viscosity may be required to prevent film rupture.

In some operations, the conditions may be more severe owing to overloads or to a combination of heavy loads and extreme shock loads, for instance, on rolling mill stands or in applications where gears are started under heavy load and/or have the capability of reversing direction. In such cases, it may be impossible to maintain an effective oil film. Hence, during a considerable part of mesh, boundary lubrication exists. This condition generally requires the use of extreme pressure (EP) oils.

Occasionally, owing to lack of space or other limiting and unavoidable factors, gears are loaded so heavily that it is difficult to maintain an effective lubricating film between the rubbing surfaces. Such a condition is quite usual for hypoid gears in the automotive field. When operating under this condition of extreme loading, the potential for metal-to-metal contact can be so severe that wear cannot be completely avoided. However, it can be controlled by the use of special extreme pressure lubricants containing additives designed to prevent welding and surface destruction under severe conditions. Only slow wear of a smooth and controlled character will then take place. Synthetic hydrocarbon lubricants formulated with extreme pressure additives have proven to be ideal lubricants for hypoid gears.

8. Drive Type

Electric motors, steam turbines, hydraulic turbines, and gas turbines are generally used in applications where the requirement is for uniform torque. Therefore, when the power transmitted by gears is developed by one of these prime movers, gear tooth loading is uniform. Reciprocating engines, however, produce variable torque, so some variation in gear tooth loading results. When gears are driven by prime movers that vary in torque, higher viscosity oils may be required to assure effective oil films. Higher viscosity oils may not be necessary when the type of drive has been considered and compensated for in the design or selection of the gear set.

9. Application Method

When lubricating oil is applied to gear teeth by means of a splash system, the formation of an oil film between the teeth is less effective than when the oil is circulated and sprayed directly on the meshing surfaces. This is particularly true of low speed, splash-lubricated units in which only a limited amount of oil may be carried to the meshing area. A higher viscosity oil is needed to offset this condition, since with higher viscosity, more oil clings to the teeth and is carried into the mesh.

When a gear set is lubricated by a pressure system rather than a splash lubrication system, there is better dissipation of heat. This is because the pressure tends to throw the oil against all internal surfaces of the gear case, and more heat is conducted away by these radiating surfaces. With a splash system, particularly a low speed unit, the oil may dribble over only a small part of the internal surface of the gear case, thus restricting heat dissipation. As a result, splash lubricated units usually run hotter and require higher viscosity oils.

10. Water Contamination

Water sometimes finds its way into the lubrication systems of enclosed gears. This water may come from cooling coils, condensed steam, washing of equipment, or condensation

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of moisture in the atmosphere. In the latter case, it is often an indication of inadequate venting of the gear case and oil reservoir. Water contamination is likely to occur in gear sets operated intermittently, with warm periods of operation alternating with cool periods of idleness causing moisture to condense. Applications in high humidity conditions where temperatures can drop to levels at or below the dew point may need to be equipped with desiccant breathers. Where moisture contamination may occur, it is necessary to use an oil with good demulsibility, that is, an oil that separates readily from water.

Water and rust also act to speed up deterioration of the oil. Water separates slowly, or not at all, from oil that has been oxidized or contaminated with dirt. In this respect, iron rust is a particularly objectionable form of contamination. Water in severely oxidized or dirty oil usually forms stable emulsions. Such emulsions may cause excessive wear of gears and bearings by reducing the lubricant's ability to provide proper lubrication and by restricting the amount of oil flowing through pipes and oil passages to the gears and bearings. Oxidized oil promotes the formation of stable emulsions, and this is another reason for using an oxidation-resistant oil in enclosed gears. Obviously then, to protect gear tooth surfaces and bearings, the oil must not only separate quickly from water when new but also must have the high chemical stability necessary to maintain a rapid rate of separation even after long service in a gear case.

11. Lubricant Leakage

Although most enclosed gear cases are oil tight, extended operation or more severe operating conditions may result in lubricant leakage at seals or joints in the casing. When the amount of leakage is high and cannot be controlled by other methods, it may be necessary to use special lubricants, such as semifluid greases, designed to resist leakage. Special considerations may be required when one is using antileak oils or semifluid greases, since these may not be consistent with the manufacturer's lubricant recommendations.

VI. LUBRICANT CHARACTERISTICS FOR ENCLOSED GEARS

The necessary characteristics of lubricants for enclosed gears may be summarized as follows.

1. Correct viscosity at operating temperature to assure distribution of oil to all rubbing surfaces and formation of protective oil films at prevailing speeds and pressures
2. Adequate low temperature fluidity to permit circulation at the lowest expected start temperature
3. Good chemical stability to minimize oxidation under conditions of high temperatures and agitation in the presence of air, and to provide long service life for the oil
4. Good demulsibility to permit rapid separation of water and protect against the formation of harmful emulsions
5. Antirust properties to protect gear and bearing surfaces from rusting in the presence of water, entrained moisture, or humid atmospheres
6. A noncorrosive nature to prevent gears and bearings from being subjected to chemical attack by the lubricant
7. Foam resistance to prevent the formation of excessive amounts of foam in reservoirs and gear cases

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8. Good compatibility with system components such as seals and paints, and with gear metallurgy

In addition to these characteristics, many modern gear sets operating under severe service conditions or in applications where loads are heavy or shock loads are present require lubricants with extreme pressure (EP) properties to minimize scuffing and destruction of gear tooth surfaces. Worm gears usually require lubricants with mild wear- and friction-reducing properties. It is important to note that some highly additized EP gear

Table 8.2 Viscosity Ranges for AGMA Lubricants

Rust and oxidation-inhibited gear oils, AGMA lube no.	Viscosity range [mm^2/s (cSt) at 40°C][a]	Equivalent ISO grade[a]	Extreme pressure gear lubricants,[b] AGMA lube no.	Synthetic gear oils,[c] AGMA lube no.
0	28.8–35.2	32		0 S
1	41.4–50.6	46		1 S
2	61.2–74.8	68	2 EP	2 S
3	90–110	100	3 EP	3 S
4	135–165	150	4 EP	4 S
5	198–242	220	5 EP	5 S
6	288–352	320	6 EP	6 S
7, 7 Comp[d]	414–506	460	7 EP	7 S
8, 8 Comp[d]	612–748	680	8 EP	8 S
8A Comp[d]	900–1100	1000	8 A EP	—
9	1350–1650	1500	9 EP	9 S
10	2880–3520	—	10 EP	10 S
11	4140–5060	—	11 EP	11 S
12	6120–7480	—	12 EP	12 S
13	190–220 cSt at 100°C (212°F)[e]	—	13 EP	13 S
Residual compounds[f] AGMA lube no.	Viscosity ranges [cSt at 100°C (212°F)][e]			
14R	428.5–857.0			
15R	857.0–1714.0			

[a] Per ISO 3448, Industrial Liquid Lubricants—Viscosity Classification. Also ASTM D 2422 and British Standards Institution B.S. 4231.

[b] Extreme pressure lubricants should be used only when recommended by the gear manufacturer.

[c] Synthetic gear oils 9S–13S are available but not yet in wide use.

[d] Compounded with 3–10% fatty or synthetic fatty oils.

[e] Viscosities of AGMA lubricant 12 and above are specified at 100°C (210°F) because measurement of viscosities of these heavy lubricants at 40°C (100°F) would not be practical.

[f] Residual compounds—diluent types, commonly known as solvent cutbacks, are heavy oils containing a volatile, nonflammable diluent for ease of application. The diluent evaporates, leaving a thick film of lubricant on the gear teeth. Viscosities listed are for the base compound without diluent.

CAUTION: These lubricants may require special handling and storage procedures. Diluent can be toxic or irritating to the skin. Do not use these lubricants without proper ventilation. Consult lubricant supplier's instructions.

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oils may have negative effects on worm gears particularly where different metallurgy such as bronze on steel, are used.

Standard 9005-D94 (ANSI/AGMA 9005-D94) of the American Gear Manufacturers Association (AGMA) combines the specifications for enclosed and open gear lubricants. This specification supersedes AGMA standard 250.04 (Lubrication of Industrial Enclosed Gearing) and 251.02 (Lubrication of Industrial Open Gearing). This AGMA standard provides specifications for rust and oxidation (R&O), compounded (included in the R&O specification), extreme pressure (EP) gear lubricants and for synthetic gear lubricants for industrial gearing. The viscosity grade ranges correspond to those in ASTM D 2422 (Standard Recommended Industrial Liquid Lubricants—ISO Viscosity Classification) and B.S. 4231 from the British Standards Institution. The AGMA specification uses gear pitch line velocities as the primary parameter for determining lubricant selection in other than double-enveloping worm gears. Earlier specifications were based on gear center distances. The AGMA grades and the corresponding ISO viscosity grades are shown in Table 8.2. At the time of this publication, AGMA was in the process of revising and releasing a new AGMA standard.

Tables 8.3 and 8.4 provide AGMA lubricant number guidelines for enclosed gearing.

Table 8.3 AGMA Lubricant Number Guidelines for Enclosed Helical, Herringbone, Straight Bevel, Spiral Bevel, and Spur Gear Drives

	AGMA lubricant numbers,[c–e] ambient temperature °C/(°F)[f,g]			
Pitch line velocity of final reduction stage[a,b]	−40 to −10 (−40 to +14)	−10 to +10 (14 to 50)	10 to 35 (50 to 95)	35 to 55 (95 to 131)
Less than 5 m/s (1000 ft/min)[h]	3S	4	6	8
5–15 m/s (1000–3000 ft/min)	3S	3	5	7
15–25 m/s (3000–5000 ft/min)	2S	2	4	6
Above 25 m/s (5000 ft/min)[h]	0S	0	2	3

[a] Special considerations may be necessary at speeds above 40 m/s (8000 ft/min). Consult gear drive manufacturer for specific recommendations.

[b] Pitch line velocity replaces center distance as the gear drive parameter for lubricant selection. The corresponding table from the previous standard is included as Annex B[i] for reference.

[c] AGMA lubricant numbers listed refer to R&O and synthetic gear oil shown in Table 4.* Physical and performance specifications are shown in Tables 1 and 3[i] EP or synthetic gear lubricants in the corresponding viscosity grades may be substituted where deemed acceptable by the gear drive manufacturer.

[d] Variations in operating conditions (surface roughness, temperature rise, loading, speed, etc.) may necessitate use of a lubricant of one grade higher or lower. Contact gear drive manufacturer for specific recommendations.

[e] Drives incorporating wet clutches or overrunning clutches as backstopping devices should be referred to the gear manufacturer as certain types of lubricant may adversely affect clutch performance.

[f] For ambient temperatures outside the ranges shown, consult the gear manufacturer.

[g] Pour point of lubricant selected should be at least 5°C (9°F) lower than the expected minimum ambient starting temperature. If the ambient starting temperature approaches lubricant pour point, oil sump heaters may be required to facilitate starting and ensure proper lubrication (see 5.1.6).*

[h] At the extreme upper and lower pitch line velocity ranges, special consideration should be given to all drive components, including bearing and seals, to ensure their proper performance.

* AGMA Table 4 is shown in Table 8.2. AGMA Tables 1 and 3, Annex B, and paragraph (5.1.6) are not included in this book.

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Table 8.4 AGMA Lubricant Number Guidelines for Enclosed Cylindrical Worm Gear Drives[a]

Pitch line velocity[b] of final reduction stage	AGMA lubricant numbers[a] ambient temperature [°C/(°F)][c,d]			
	−40 to −10 (−40 to +14)	−10 to +10 (14 to 50)	10 − 35 (50 to 95)	35 − 55 (95 to 131)
<2.5 m/s (450 ft/min)	5S	7 Comp	8 Comp	8S
>2.5 m/s (450 ft/min)	5S	7 Comp	7 Comp	7S

[a] AGMA lubricant numbers listed above refer to compounded R&O oils and synthetic oils shown in Table 4.* Physical and performance specifications are shown in Tables 1 and 3.† Worm gear drives may also operate satisfactorily using other types of oil. Such oils should be used, however, only with approval of the gear manufacturer.

[b] Pitch line velocity replaces center distance as the gear drive parameter for lubricant selection.

[c] Pour point of the oil used should be at least 5°C (9°F) lower than the minimum ambient temperature expected.

[d] Worm gear applications involving temperatures outside the limits shown, or speeds exceeding 2400 rpm or 10 m/s (2000 ft/min) sliding velocity, should be referred to the manufacturer. In general, for higher speeds a pressurized lubrication system is required along with adjustments in recommended viscosity grade.

* AGMA Table 4 is shown in Table 8.2.

† AGMA Tables 1 and 3 are not included in this book.

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A. Factors Affecting Lubrication of Open Gears

In contrast to enclosed gears that are flood-lubricated by splash or circulation systems, there are many gears for which it is not practical or economical to provide oil-tight housings. These so-called open gears can only be sparingly lubricated and perhaps only at infrequent intervals. Gears of this type are lubricated by either a continuous or an intermittent method. Some of the more sophisticated open gearing systems may be lubricated with a full circulation system that captures and filters the oil for reuse.

The three most common continuous methods are splash, idler gear immersion, and pressure. In the first two the lubricant is lifted from a reservoir or sump (sometimes referred to as a slush pan) by the partially submerged gear or an idler. Pressure systems require a shaft or independently driven pump to draw oil from a sump and spray it over the gear teeth. With the continuous methods of application, lubrication of open gears is similar to that with enclosed gears. Since the gears are usually large and relatively slow moving, very high viscosity lubricants are required. Most gears lubricated in any of these ways are equipped with relatively oil tight enclosures. AGMA has published guideline for continuous methods of application (Table 8.5).

In addition, many different intermittent methods of application are used. Some are arranged for automatic timing, while others must be controlled manually. Methods used include automatic spray, semiautomatic spray, forced-feed lubricators, gravity or forced drip, and hand application by brush. Grease-type lubricants can be applied by means of hand or power grease guns or by a centralized lubrication system.

With the intermittent methods of application, fluid films may exist when lubricant is first applied to the gears. However, these films quickly become thinner as the lubricant is squeezed aside, whereupon only extremely thin films remain on the metal surfaces. During much of the time, therefore, these gears operate under conditions of boundary

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Table 8.5 AGMA Lubrication Number Guidelines for Open Gearing (continuous method of application)[a,b]

		Pitch line velocity				
		Pressure lubrication		Splash lubrication		Idler immersion
Ambient temperature [°C (°F)][c]	Character of operation	<5 m/s (1000 ft/min)	>5 m/s (1000 ft/min)	<5 m/s (1000 ft/min)	5–10 m/s (1000–3000 ft/min)	≤1.5 m/s (300 ft/min)
−10 to 15[d] (15–60)	Continuous	5 or 5 EP	4 or 4 EP	5 or 5 EP	4 or 4 EP	8–9 8 EP–9 EP
	Reversing or frequent ''start–stop''	5 or 5 EP	4 or 4 EP	7 or 7 EP	6 or 6 EP	8–9 8 EP–9 EP
10 to 50[d] (50–125)	Continuous	7 or 7 EP	6 or 6 EP	7 or 7 EP	8–9[e] 8 EP–9 EP	11 or 11 EP
	Reversing or frequent ''start–stop''	7 or 7 EP	6 or 6 EP	9–10[f] 8 EP–9 EP		

[a] AGMA lubricant numbers listed refer to gear lubricants shown in Table 4.* Physical performance specifications are shown in Tables 1 and 2.[g] Although both R&O and EP oils are listed, the EP is preferred. Synthetic oils in the corresponding viscosity grades may be substituted where deemed acceptable by the gear manufacturer.
[b] Does not apply to worm gearing.
[c] Temperature in vicinity of the operating gears.
[d] When ambient temperatures approach the lower end of the given range, lubrication systems must be equipped with suitable heating units for proper circulation of lubricant and prevention of channeling. Check with lubricant and pump suppliers.
[e] When ambient temperature remains between 30°C (90°F) and 50°C (125°F) at all times, use 9 or 9 EP.
[f] When ambient temperature remains between 30°C (90°F) and 50°C (125°F) at all times, use 10 or 10 EP.
* AGMA Table 4 is shown in Table 8.2. AGMA Tables 1 and 2 are not included in this book.
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lubrication. Under boundary conditions, the extremely thin film must resist being rubbed or squeezed off the surfaces of the teeth. The ability of the film to resist the rubbing action depends both on its viscosity and on the action that takes place between the lubricant and the metal surfaces. The lubricating film must bond to the tooth surfaces strongly enough to minimize metal-to-metal contact (and resultant wear). With a properly selected lubricant applied at sufficiently frequent intervals, wear may be kept to a negligible amount.

1. Temperature

Whether it is possible to maintain adequate boundary films under the pressures existing between meshing teeth depends, among other things, on the lubricant characteristics and the operating temperature to which the gears are exposed. Heat causes the lubricant to thin out and drop off or be thrown off the gear teeth, decreasing the amount of lubricant remaining on the rubbing surfaces. This results in thinner films. Increasing temperatures also decrease the resistance of the lubricant to stay on the contact surfaces. Thus, high temperatures require more viscous lubricants. Conversely, when gears operate at low temperatures, the lubricant must not become so viscous or hard that it will not distribute

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properly over the tooth surfaces. At the same time, the lubricant must be such that it does not harden and chip or peel from the teeth at the lowest temperatures encountered.

2. Dust and Dirt

Many open gears, whether operating outdoors or indoors, are exposed to dusty and dirty conditions. Abrasive dust, adhering to oil wetted surfaces, will form a lapping compound that causes excessive wear of the teeth. When viscous lubricants are used, the dirt may pack in the clearance space at the roots of the teeth, forming hard deposits. Packed deposits between gear teeth tend to spread the gears and overload the bearings. These deposits that build up in the tooth root area, if hard enough, can also lead to tooth wear and possible breakage.

3. Water

Open gears operating outdoors are often exposed to rain or snow, and outdoor and indoor gears contact the splash of process fluids. To protect gear tooth surfaces against wear or rust and corrosion, the lubricant must resist being washed off the gears by these fluids.

4. Method of Application

The method of application must be considered when one is selecting a lubricant for open gears. If the lubricant is to be applied by drip force-feed lubricator, or spray, it must be sufficiently fluid to flow through the application equipment. For brush application, the lubricant must be sufficient fluid to be brushed evenly on the teeth. In any case, during operation, the lubricant should be viscous and tacky to resist squeezing from the gear teeth. Very viscous lubricants of some types can be thinned for application by heating, or diluent-type products may be used. These latter products contain a nonflammable diluent that reduces the viscosity sufficiently for application. Shortly after application, the diluent evaporates, leaving the film of viscous base lubricant to protect the gear teeth. When open gears are lubricated by dipping into a slush pan, the lubricant must not be so heavy that it channels as the gear teeth dip into it. When open gears are lubricated by grease-type materials, the consistency and pumpability must permit easy application under the ambient conditions prevailing. Lubricant quantity guidelines for intermittent methods of application can be found in publication 9005-D94 from ANSI/AGMA (Table 8.6).

5. Load Characteristics

Open gears are often heavily loaded and shock loads may be present. These conditions usually require the use of lubricants with enhanced antiwear and EP properties.

VII. AGMA SPECIFICATIONS FOR LUBRICANTS FOR OPEN GEARING

AGMA Standard 9005-D94 also contains specifications for open gear applications. The specifications include R&O gear oils (compounded gear oils are included in this specification), EP gear oils, and synthetic gear oils. AGMA does publish specifications for *residual gear compounds* for open gearing included in Tables 8.2 and 8.6. Generally heavier grades of straight mineral oils and EP oils with viscosities ranging from 400 to 2000 cSt at 100°C (without diluent) are considered to be residual compounds. Other heavy-bodied lubricants designed for open gear applications would be included in this category. These are generally mixed with diluent for ease of application.

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Table 8.6 AGMA Lubricant Number Guidelines for Open Gearing Intermittent Applications[a–c] [where gear pitch line velocity does not exceed 7.5 m/s (1500 ft/min)]

	Intermittent spray systems[e]			Gravity feed or forced drip method[f]	
Ambient temperature [°C (°F)][d]	R&O or EP lubricant	Synthetic lubricant	Residual compound[g]	R&O or EP lubricant	Synthetic lubricant
−10 to 15 (15–60)	11 or 11 EP	11 S	14 R	11 or 11 EP	11 S
5 to 40 (40–100)	12 or 12 EP	12 S	15 R	12 or 12 EP	12 S
20 to 50 (70–125)	13 or 13 EP	13 S	15 R	13 or 13 EP	13 S

[a] AGMA Viscosity number guidelines listed refers to gear oils shown in Table 4.*
[b] Does not apply to worm gearing.
[c] Feeder must be capable of handling lubricant selected.
[d] Ambient temperature is temperature in vicinity of the gears.
[e] Special compounds and certain greases are sometimes used in mechanical spray systems to lubricate open gearing. Consult gear manufacturer and spray system manufacturer before proceeding.
[f] EP oils are preferred, but may not be available in some grades.
[g] Diluents must be used to facilitate flow through applicators.
* AGMA Table 4 is shown in Table 8.2.
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VIII. CYLINDERS

Cylinders are found in reciprocating compressors, reciprocating internal combustion engines, some older steam engines, hydraulic systems, some air tools, and linear actuators. For the purposes of this brief discussion of cylinder lubrication, reference is made only to compressors and internal combustion engines.

In understanding cylinder lubrication of reciprocating equipment such as compressors or internal combustion engines, pistons, and piston rings are an integral part. In engines and compressors, the pistons are the main component that is used to compress air or other gases. Resultant pressures and temperatures can be high, and therefore some method of sealing the compressed media from bypassing the piston and cylinder is needed. This sealing is provided by the rings and the lubricant. The rings also serve to maintain alignment of the piston in the cylinder bore, act as the bearing surface for controlling wear, and control the oil films on the cylinder walls.

The lubrication of cylinders involves conditions that range from boundary films to full fluid films. For example, at the top and bottom or piston travel in the cylinder, the piston and rings actually stop and reverse direction.* These areas are called *ring reversal areas.* In these areas, the full fluid films developed by the rings in motion are lost and boundary lubrication conditions exist until the motion of the pistons and rings reestablishes the films. In the ring reversal area at the bottom of travel, the reestablishing of the films

* Engines and compressors can have vertical, horizontal, or ''V'' configurations. Some reciprocating engines also have opposed pistons (i.e., an upper and lower piston in each cylinder). For discussion purposes, vertical travel of pistons has been selected, but it refers to the outward-most position in horizontal or other angles of the piston travel. Discussion also is limited to single-acting compressor cylinders.

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is not as critical as at the top of travel (assuming vertical travel). At the top of travel, pressures and temperatures are much higher, and as a result, most of the wear in this type of equipment occurs at the top ring reversal area.

The lubricant's primary function is to reduce wear and provide sealing. In addition, the lubricant must minimize formation of deposits, provide protection against rust and corrosion, and handle moisture and other liquids and gases entering the cylinders as a result of compression or combustion. In the compression of some gases, the solubility of the gases in the lubricant films may result in decreasing the oil's viscosity. This may dictate use of higher viscosity lubricants. Depending on application and operating conditions, the selection of a correct lubricant can be complex. This selection process is discussed in greater detail in Chapter 10 (Internal Combustion Engines) and Chapter 17 (Compressors).

IX. FLEXIBLE COUPLINGS

When two rotating shafts are to be connected, some degree of misalignment is almost unavoidable. This is either because of static effects such as deflection of the shafts or thermal effects causing the shafts to change relative positions in their supporting bearings. Misalignment may be angular where the shafts meet at an angle that is not 180°, parallel where the shafts are parallel but displaced laterally, or axial, as results, for example, from endplay. In some cases, all three types of misalignment may be present. To accommodate misalignment, flexible couplings of various types are used to connect shafts together.

In addition to protecting the machines against stresses resulting from misalignment, flexible couplings transmit the torque from the driving shaft to the driven shaft and may help to absorb shock loads.

Universal joints and constant velocity joints may properly be considered as flexible couplings. Both types of joint will accommodate relatively large amounts of misalignment, and are used to some extent in industrial applications; see Chapter 16 for a detailed discussion. Here we consider only types of flexible coupling used in industrial applications, where the amount of misalignment is relatively small.

Several types of lubricated flexible coupling are in use. Gear-type couplings (Figure 8.56) have hubs with external gear teeth (or splines) keyed to the shafts. A shell or sleeve with internal gear teeth at each end meshes with the teeth on the hubs and transmits the

Figure 8.56 Typical Falk gear coupling components.

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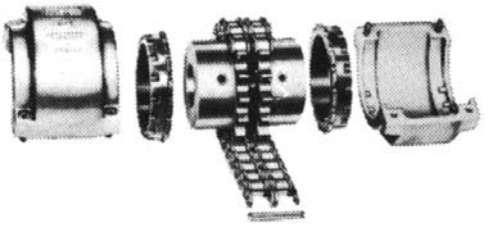

Figure 8.57 Chain coupling with cover.

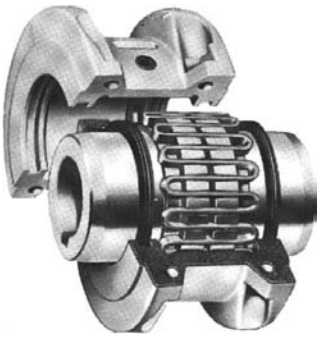

Figure 8.58 Spring coupling, horizontally split cover.

torque from the driving shaft to the driven shaft. Misalignment is accommodated in the meshing gears. Where large amounts of misalignment must be accommodated, special tooth forms that permit greater angular motion are used.

Chain couplings (Figure 8.57) have sprockets connected by a chain wrapped around and joined at the ends. The chain may be of either the roller or silent type. Roller chain couplings depend on the relative motion between the rollers and sprockets for their flexibility. Barrel-shaped rollers and special tooth forms may be used where large amounts of misalignment must be accommodated. Silent chain couplings depend on the clearance between the chain links and sprocket teeth for their flexibility.

Spring-laced couplings (Figure 8.58), also called steel grid or spring-type couplings, are made of two serrated flanges laced together with metal strips. Flexibility is obtained from movement of the metal strips in the serrations.

There are two types of sliding couplings. Sliding block couplings have two C-shaped jaw members connected by a square center member. The jaws are free to slide on the surfaces of the block to accommodate misalignment. Sliding disk couplings have flanged hubs with slots machined in the flanges. These slots engage jaw projections on a center disk. The floating center disk allows relative sliding movement between the flange slots and the disk jaws.

Slipper-type couplings, used mainly on rolling mills, are actually a type of universal joint. A C-shaped jaw attached to one shaft is fitted with pads that can turn in their recesses. A tongue on the other shaft fits between the pads. The tongue can move sideways in the pads to accommodate lateral misalignment, and the pads can rotate in their recesses to accommodate angular misalignment.

Flexing member couplings compensate for misalignment both by flexing of the components and by sliding between them. The radial spoke coupling uses thin steel laminations to connect the coupling flanges. Clearance in the slots of the outer flange and flexing of the laminations allows for misalignment. Axial spoke couplings have flexible pins mounted in sintered bronze bushings connecting the flanges. The pins can flex or move in and out in their bushings to accommodate misalignment.

A. Lubrication of Flexible Couplings

All motion in flexible couplings that require lubrication is of the sliding type. Motion is more or less continuous when a coupling is revolving, but the amount of motion may be

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Table 8.7 Grease Lubricated Coupling Operating Classifications[a]

Operating conditions	Operating groups I	II	III
1. Rotational speed (rpm)			
d (in.)	≤ 3600	$\geq 2800^{b}/d^{1/2}$	$\leq 2800^{b}/d^{1/2}$
d (mm)	≤ 3600	$\geq 14{,}100/d^{1/2}$	$\leq 14{,}100/d^{1/2}$
2. Misalignment (degree)	≤ 0.75	≤ 0.5	≥ 0.75
3. Continuous torque			
T (lb-in.)	$\leq 1200d^3$	$\geq 1200d^{3\ c}$	$\geq 1200d^{3\ c}$
T (N·m)	$1200d^3/8.8\ (25.4)^3$	$\geq 8.3 \times 10^{-3}d^3$	$\geq 8.3 \times 10^{-3}d^3$
4. Peak torque	$\leq 2.5\ T$	$\leq 2.5\ T$	$\geq 2.5\ T$
5. Maximum coupling	150°F	170°F	212°F
Surface temperature	(65°C)	(77°C)	(100°C)
6. Normal relube	6–12	12–36	1 or less
Interval (months)[d]			

[a] Definitions: d, shaft diameter; T, torque; G, ratio of actual acceleration to gravitational acceleration [$G \approx 28.4d$ (rpm) $^2/10^6$ when d is in inches, $1.12d$ (rpm) $^2/10^6$ when d is in millimeters]; d is not an exact value because it relates approximately to its pitch radius of a coupling.

[b] Relates to centrifugal force on the lubricant of approximately 200 g.

[c] Relates to shaft torsional stress of approximately 6000 lb/in.^{2} (0.207 MPa).

[d] The actual relube interval is dependent on experience with the specific application.

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small, varying with the amount of misalignment between the shafts. Contact pressure may be high, particularly when there is considerable misalignment, since misalignment tends to reduce the surface area over which the load is distributed. The thickness of the lubricating films formed and the type of lubricant required depend to a considerable extent on the type of coupling.

AGMA defines three major operating groups for grease-lubricated flexible couplings as a function of shaft diameter, rotational speed, misalignment, torque, and coupling surface temperature. For the three operating groups, three coupling grease specifications are defined: AGMA CG-1, CG-2, and CG-3. These specifications are defined in AGMA Standard 9001-A86 (Table 8.7), as are AGMA flexible coupling operating classifications and grease specifications (Table 8.8).

1. Gear Couplings

Gear couplings can transmit more torque than any other type of flexible coupling of equal size. Since they rotate continuously, a supply of lubricant is usually contained in the assembly. As the unit revolves, centrifugal force causes the lubricant to be thrown outward, forming an annulus of lubricant that should completely submerge the gear teeth. The relative motion between the gear teeth may be sufficient to create and maintain appreciably thick lubricating films.

Either oil or grease may be used for gear couplings. Greases are often used where speeds are normal to high and temperatures are moderate. For normal speeds, a soft grease

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Table 8.8 AGMA Coupling Grease Specifications

Characteristic (test method)[a]	Type CG-1	Type CG-2	Type CG-3
Minimum base oil viscosity:			
In centistokes	198 at 40°C (104°F)	288 at 40°C (104°F)	30 at 100°C (212°F)
In SSU (approx.)	900 at 100°F (38°C)	1300 at 100°F (38°C)	150 at 210°F (99°C)
Separation characteristics[b]	K36 ≤ 60/24, or 8% maximum fluid insoluble material	K36 ≤ 24/24	No restriction
National Lubrication Grease Institute (NLGI) grade			
a) metallic grid	1–3	1–3	1–3
b) Gear or chain			
where rpm ≥ $200/\sqrt{d}$ in $1008/\sqrt{d}$ mm	0–3	0–1	1–2
where rpm ≤ $200/\sqrt{d}$ in $1008/\sqrt{d}$ mm[c]	0–1	Not applicable	1–2
Minimum dropping point	190°F (88°C)	195°F (91°C)	302°F (150°C)
Compatibility[d]	The coupling grease must be compatible with coupling seals and gaskets		
Oxidation resistance—max pressure drop at 100 h	20 lb/in.^{2} (13,790 Pa)	20 lb/in.^{2} (13,790 Pa)	20 lb/in.^{2} (13,790 Pa)
Antirust properties	Not required	ASTM rating pass	ASTM rating pass
Antiwear additives[d]	Not required	Not required	Required[e]
Extreme pressure (EP) additives[d]	Not required[f]	Not required[f]	Required
Timken OK load	Not required	Not required	40 lb minimum
Four-ball EP test	Not required	Not required	Weld point 250 kg minimum

[a] Accepted test methods;
Viscosity, ASTM D 445
Grease composition, ASTM D 128
Centrifuge test, ASTM D 4425
NLGI grade, ASTM D 217
Dropping point, ASTM D 566 or D 2265
Antirust properties, ASTM D 1743
Oxidation resistance, ASTM D 942
Four-ball EP test, ASTM D 2596
Timken OK load, ASTM D 2509.

[b] ASTM centrifuge test.

[c] Relates to a centrifugal force on the lubricant of approximately 10 *g*.

[d] No test method.

[e] Experience has shown that a minimum of 5% (by weight) MoS_2 (molybdenum disulfide) is beneficial for couplings with hardened teeth.

[f] EP additives recommended by some coupling manufacturers.

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made with a high viscosity oil, often with extreme pressure properties, is usually used. For higher speeds, a stiffer grease may be required (see Tables 8.7 and 8.8).

2. Chain Couplings

Chain couplings (see Figure 8.57) are usually grease-lubricated. Couplings without dust covers are generally lubricated by brushing with grease periodically. The grease must resist throwoff and must penetrate the chain joints to provide lubrication. Generally, soft greases with good adhesive properties are required. Couplings with covers are packed with grease, usually of a type similar to those used in geared couplings.

3. Spring-Laced Couplings

Couplings like those shown in Figure 8.58 are grease-packed. Again, greases similar to those used in geared couplings are usually required.

4. Sliding Couplings

Sliding block couplings may have the block formed from an oil-impregnated, sintered metal. In other cases the block contains a reservoir. In the latter case, high viscosity oils with good adhesiveness are required to minimize throwoff. Sliding disk couplings require similar oils.

5. Slipper Joint Couplings

Slipper joint couplings are usually large and carry both heavy loads and shock loads, so the lubrication requirements are severe. All-loss systems are usually used to apply the lubricant. High viscosity oils with enhanced film strength are required.

6. Flexible Member Couplings

Axial spoke couplings have prelubricated bushings for the pins. When such couplings are enclosed in a dust cover, a soft, adhesive grease is required for packing. Radial spoke couplings require high viscosity oil or semifluid grease.

7. Lubrication Techniques

The preferred method of lubrication is to manually pack flexible couplings before closing. This procedure should include a coating of grease on all working surfaces, including seals.

To lubricate a coupling that is assembled, remove the highest lube plug. Slowly pump the quantity of grease specified by the coupling manufacturer via the lowest lube plug. It is preferred that the coupling be rotated so that the higher lube plugs are located at three and nine o'clock to assure adequate filling of the bottom half of the coupling. All working surfaces including seals should be coated with grease.

Key fits and keyways should be sealed with an oil-resistant sealing compound.

The proper quantity of grease is specified by the manufacturer, since couplings of varying series and shaft diameter require different amounts. If in doubt, the coupling should not be filled more than 75% of capacity to allow for thermal expansion.

Seals that become damaged or distorted should be replaced to avoid premature coupling failure from lack of lubrication.

Oil is generally preferred for gear couplings operating at normal to high speeds and high temperatures when the coupling is large enough to hold a reasonable supply of oil. Couplings may be oil-filled, in which case they require only enough oil to cover the gear teeth during operation. Continuously supplied couplings may be designed to recirculate

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the oil, or to collect it to permit drainage and disposal after use. Continuous supply systems generally are required where operating temperatures are high. Regardless of the method of applying the oil, quite high viscosity oils with good antiwear properties are required.

X. DRIVE CHAINS

Drive chains are used to transmit power in a wide variety of applications. Chains fall into two general categories:

1. Machined surface chains suitable for high speed, precision drives
2. Cast or forged link chains, usually made without machined surfaces and suitable for lower speed and power or where environmental conditions dictate the use of low cost drives.

A. Silent and Roller Chains

Silent and roller chains are precision mechanisms that, when properly lubricated, will transmit high power at high speeds and will give long service life. Silent chains, also called inverted tooth chains, operate in such a manner that the links of the chain almost completely fill the clearance space in the sprocket throughout the arc of contact, greatly reducing backlash and minimizing noise. Roller chains are available in various designs, and with 20 strands or more for use where high powers must be transmitted.

With silent chains, the lubricant must penetrate between the links and distribute along the seat pins to provide lubricating films on the rubbing surfaces. With roller chains, the lubricant must coat the outsides of the rollers to lubricate the contact surfaces of the rollers and sprockets and must also penetrate and distribute along the rubbing surfaces between the rollers and bushings. As a result, in both types of chain, the viscosity of the lubricating oil used is extremely important. Oils with enhanced film strength are also generally desirable, both to improve protection against wear and to permit the use of lighter oils with better penetrating properties.

Silent and roller chains usually are installed in the oil-tight housings, where the lubricant is applied by force feed, oil mist, slinger, or oil ring, or by the dipping of the chain or sprocket into a bath of oil. Occasionally these chains are used without housings for low speed, low power drives. For applications of this type, the lubricant is applied by one of the all-loss methods such as drip, wick feed oiler, brush, oil mist, aerosol, or pour can. The characteristics of the method of application must be taken into account in lubricant selection, as well as the requirement for controlling drip and throwoff when the chain is not enclosed.

Normally drive chains of these types are not grease-lubricated. Under certain environmental conditions, such as corrosive atmospheres, a properly selected grease may provide better protection than oil. If maintaining proper lubrication is difficult because of chain location or if excess lubricant is frequently thrown off, a lube-free chain should be considered when replacement becomes necessary.

B. Cast or Forged Link Chains

Detachable link, pintle, fabricated, and cast roller types are among the chain forms available. Detachable link chains may be of cast malleable iron, or medium carbon steel hardened to improve strength and wear characteristics. Pintle links may be either cast or forged.

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Table 8.9 Viscosity of Oil for Silent and Roller Chains (guideline)

Ambient temperature			
°F	°C	Recommended viscosity	ISO viscosity grade
−20 to +20	−30 to −7	SAE 10	46
20 to 40	−7 to −4	SAE 20	68
40 to 100	4 to 38	SAE 30	100
100 to 120	38 to 49	SAE 40	150
120 to 140	49 to 60	SAE 50	220

Source: American Chain Association.

These chains usually operate at low speeds, transmitting relatively low power; although fabricated chains suitable for speeds up to 1000 ft (300 m) per minute are available. The chains may be open, or fitted with dust covers. Under most conditions, they require viscous, tacky oils, but under dusty or dirty conditions lower viscosity oils may reduce dirt pickup. Occasionally, diluent-type open gear lubricants that have a dry, rubbery surface after application may provide better performance.

C. Viscosity Selection

The American Chain Association guidelines for lubricant viscosities for silent and roller chains are shown in Table 8.9. Where drip or throwoff may be a problem, higher viscosity oils or oils with special adhesive properties may be desirable. Where water washing occurs, oils compounded to resist water washing and protect against rusting are required.

XI. CAMS AND CAM FOLLOWERS

In many machines, rotary motion must be converted to linear motion. For linear motion that is comparatively short, this conversion is commonly accomplished with a cam and cam follower combination (cam follower is sometimes referred to as a tappet). Probably the best-known applications of these mechanisms is for operating intake and exhaust valves in a reciprocating internal combustion engine, but many other applications also exist. Since the cam will lift only the cam follower, some arrangement must be made to keep the cam follower in contact with the cam. This may be accomplished by transmitting load from the mechanism being actuated, or directly with a loading mechanism such as a return spring.

There are three general types of cam follower, to be selected according to such factors as loads, speeds, and the complexity of the cam shape.

1. Flat surface cam followers are used in valve systems of most older and quite a few newer internal combustion engines. To conserve fuel, many newer automotive internal combustion engines are using roller tappet camshafts designs.
2. Roller cam followers may be used when loads are extremely high when reduced frictional characteristics are desired.
3. Spherical cam followers may be used where precise conformity to a complex cam is required.

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Of these, the roller generally has the least severe lubrication requirements, since the rolling contact tends to develop and maintain films.

In many cases with flat cam followers, the cam lobe is tapered and the cylindrical cam follower, which is free to rotate, is located with one axis toward one side of the cam. With this arrangement, the cam follower will be rotated as it is forced upward by the cam. This spreads the wear more uniformly over the surface of the cam follower and may promote the formation of oil films.

In heavily loaded applications, both flat and spherical cam followers operate on EHL films at least part of the time. Even then, lubricants with antiwear additives are often necessary if rapid wear and surface distress are to be avoided. In the case of four-cycle gasoline engines, for example, one of the functions of the oil additive zinc dithiophosphate, now used almost universally, is to provide antiwear activity for the camshaft and valve lifters (cam followers). With the increased use of roller follower cams in automotive-type engines, the requirements for antiwear additives may be changed, mainly to prolong life of emission control devices such as catalytic converters by reducing the phosphorus levels in oils.

Most cam and cam follower machine elements are included in lubrication systems supplying other machine elements as well. For this reason, lubricants are rarely selected to meet the requirements of cams and cam followers alone. Generally, the lubricant is selected to meet the basic requirements of the system, and if special requirements exist for the lubrication of cams and followers, as in the case of a gasoline automobile engine, these must be superimposed on the basic requirements.

XII. WIRE ROPES

Wire rope or cable is used for a great variety of purposes ranging from standing (stationary) service, such as guys or stays and suspension cables, to work involving drawing or hoisting heavy loads. In these various services, all degrees of exposure to environmental conditions are encountered. These range from clean, dry conditions in applications such as elevator cables in office buildings, to full exposure to the elements on outdoor equipment, immersion in water (e.g., on dredging equipment), and exposure to corrosive environments such as acid water found in many mining applications. These and other operating factors require that wire ropes be properly lubricated to provide long rope life and maximum protection against rope failure where the safety of people is at stake.

A wire rope consists of several strands laid (helically bent, not twisted) around a core. The core is usually a rope made of hemp or other fiber, but an independent wire rope or strand may be used. Each strand consists of several wires laid around the core, which usually consists of one or more wires but may be a small fiber rope. The number of wires per strand typically ranges from 7 to 37 or more.

A. Need for Lubrication

A number of factors contribute to the need for proper lubrication of wire ropes.

1. Wear

Each wire of a wire rope can be in contact with three or more wires over its entire length. Each contact is theoretically along a line, but this line actually widens to become a narrow band because of a deformation under load. As load is applied and as a rope bends or

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flexes over rollers, sheaves, or drums, stresses are set up that cause the strands and individual wires to move with respect to each other under high contact pressures. Unless lubricating films are maintained in the contact areas, considerable friction and wear result from these movements.

2. Fatigue

One of the principal causes of wire rope failures is metal fatigue. Bending and tension stresses, repeated many times, causes fatigue. Eventually, individual wires break and the rope is progressively weakened to the extent that it must be removed from service. If lubrication is inadequate, the stresses are increased by high frictional resistance to the movement of the wires over one another, fatigue failures occur more rapidly, and rope life is shortened.

3. Corrosion

Another principal cause of rope failure is corrosion. This covers both direct attack by corrosive materials, such as acid water encountered in mines, and various forms of rusting. To protect against corrosion, lubricant films that resist displacement by water must be maintained on all wire surfaces.

4. Core Protection

Wear, deterioration, or drying out of the core result in reduction of the core diameter and loss of support for the strands. The strands then tend to overlap, and severe cutting or nicking of the wires may occur. The lubricant applied in service must be of a type that will penetrate through the strands to the core to minimize friction and wear at the core surface, seal the core against water, and keep it soft and flexible.

B. Lubrication in Manufacture

During manufacture, wire rope cores are saturated with lubricant. A second lubricant, designed to provide a very tenacious film, is usually applied to the wires and strands to lubricate and protect the wires and to help keep (seal) the lubricant in the core as they are laid up. These lubricants protect the rope during shipment, storage, and installation.

C. Lubrication in Service

Much of the core lubricant applied during manufacture is squeezed out when the strands are laid, and additional lubricant is lost from both the core and strands as soon as load is applied to a rope. As a result, in-service lubrication must be started almost immediately after a rope is placed in service.

Proper lubrication of wire ropes in service is not easy to accomplish. Some of the types of lubricant required for wire ropes may not be easy to apply, and often wire ropes are somewhat inaccessible. Various methods of applying lubricants are used, including brushing, spraying, pouring on a running section of the rope, reliance on drip or force-feed applicators, and running the rope through a trough or bath of lubricant. Generally, the method of application is a function of the type of lubricant required to protect a rope under the conditions to which it is exposed.

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D. Lubricant Characteristics

Wire rope lubricants should be able to do the following:

1. Form a durable, adhesive coating that will not be thrown or wiped from the rope as it operates over pulleys or drums
2. Penetrate between adjacent wires, to lubricate and protect them against wear and to keep the rope core from drying out and deteriorating
3. Provide lubrication between pulleys and sheaves and the rope
4. Resist being washed off by water
5. Protect against rusting or corrosion by acid, alkaline, or salt water
6. Form nonsticky films so that dust and dirt will not build up on ropes
7. Remain pliable and resist stripping at the lowest temperatures to which the rope will be exposed
8. Resist softening or thinning out to the extent that throwoff or drippage occurs at the highest temperatures at which the rope will operate
9. Be suitable for application to the rope under the service conditions encountered

These requirements necessitate some compromises. Wire rope lubricants may be formulated with asphaltic or petrolatum bases and usually contain rust preventives and materials to promote metal wetting and penetration. Diluent products are used in some cases for ease of application. Grease products containing solid lubricants such as graphite or molybdenum disulfide are also used.

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